OCT 30 1924

# MECHANICAL ENGINEERING

INCLUDING THE ENGINEERING INDEX

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ARTHUR D. LITTLE

*(From address at Franklin Institute Centenary, September 19, 1924)*

NOVEMBER 1924

THE MONTHLY JOURNAL PUBLISHED BY THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

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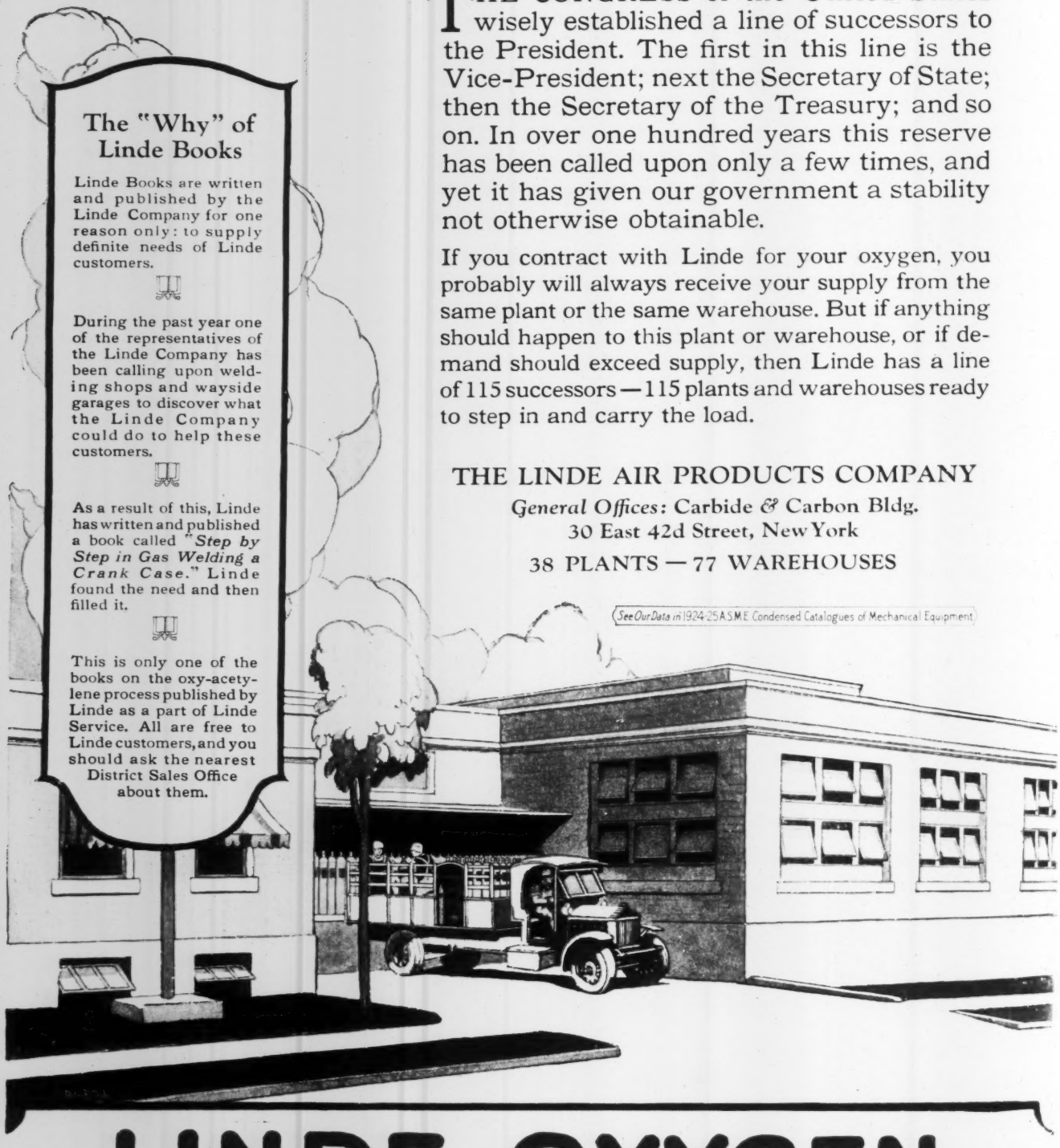

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Volume 46 November, 1924 Number 11

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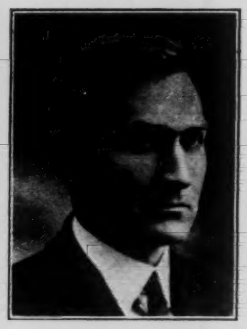

A. B. Cox

E. E. MacMorland

N. C. Barnard

F. J. Pancratz

# Contributors to this Issue

**Dr. Heinrich Zoelly,** author of the leading paper in this issue on The Zoelly Turbine-Driven Locomotive, is chairman of the Board of Directors of Escher Wyss & Co., Zurich, Switzerland, and was for many years, chief engineer and a director of that firm. After a number of years of specialization in the design and construction of hydraulic machinery he began about twenty-eight years ago to experiment with the steam turbine.

The steam turbine which bears his name and which is now so widely adopted, was first brought out by Escher Wyss & Co. in 1903. In its development Dr. Zoelly was guided not only by his own extensive experience but by that of his firm in the development of the water turbine, particularly of the impulse type.

* * * * *

**Norris C. Barnard,** whose paper on Die Casting is included in this issue, has been connected with the Doehler Die Casting Co., as draftsman, checker, plant engineer of the Chicago plant, and as assistant to chief engineer of the Brooklyn plant, Brooklyn, N. Y. Mr. Barnard was graduated from high school in Brooklyn and then attended Swarthmore College, from which he was graduated in 1919 with the degree of A.B. In 1923 he received his M.E., also from Swarthmore. At present he is associated with the Boston Pressed Metal Co., Worcester, Mass.

* * * * *

**F. J. Pancratz,** associated with the McAlear Manufacturing Co., Chicago, Ill., is the author of the article on Wind Power for Farm Electric Plants. He was graduated in 1908 from the University of Minnesota with the degree of E.E., and served part of the apprenticeship course for college men with the Westinghouse Electrical & Manufacturing Co. He was then engaged by the Central Colorado Power Co., Denver, Colo. Later he was with the Pacific Gas & Electric Co., San Francisco, Cal., and then became chief electrician of the Federal Mining & Smelting Co., Wallace, Idaho. Since 1914 Mr. Pancratz has been connected with several large manufacturing companies in Chicago in research work and the design of electrical apparatus, automatic machinery, and pulverized-coal equipment. Upon resigning from the engineering department of the Raymond Brothers' Engineering Co., he became associated with the McAlear Manufacturing Co., of Chicago.

* * * * *

**E. L. Thearle,** author of the paper on The Rotating Disk, is an instructor in the mechanics of engineering at Cornell University. Mr. Thearle was born in Chicago, Ill. He received his early education in the schools of southern California, and then attended the California School of Mechanical Arts in San Francisco. For several years he was employed in San Diego and San Francisco as a draftsman on the design of gasoline engines and marine steam engines. Mr. Thearle entered Cornell University in 1917 and was appointed to an instructorship of mechanics in 1920. In 1923 he received his degree of M.E.

* * * * *

**Capt. E. E. MacMorland,** who contributes a paper on The Engineer in Industrial Mobilization Planning, was educated in Missouri and Princeton Universities. He holds the degrees of A.B. and A.M. In 1916 he entered the Coast Artillery Corps of the Army and after service at Fort Monroe, Va., and the Presidio, San Francisco, Cal., went to France with the 62d Artillery. He was transferred from France to North Russia and later worked with the Peace Commission in the Baltic States, returning to the United States in 1920. He was then sent for further service in the Coast Artillery, had a short tour of duty with the Transportation Service of the Quartermaster Corps in 1921, and was finally transferred to the Ordnance Department in May, 1921. Since the latter part of that year Captain MacMorland has been on duty in the office of the Chief of Ordnance and has been connected with the administration of the District System since its inception.

He holds an A. E. F. citation for meritorious service and received the Distinguished Service Order (British) for conspicuous service in command of an American Railway Battalion in the Murmansk Sector, North Russian theater of operations.

* * * * *

**Anthony Bruce Cox,** who writes in this issue on Limiting Cases in Involute Spur Gearing, is design draftsman in a branch of the Navy Department of the U. S. Government, located at the East Pittsburgh works of the Westinghouse Electric & Manufacturing Co. Mr. Cox was graduated from the engineering course at Purdue University in 1911 and then followed six years' experience covering electrical installation and operation, electrochemical plant building and apparatus design and operation, steam-power plant operation, and machine design. At the end of this period he became connected with the Navy Department. It was in this latter position while investigating the causes of noisy gearing in certain cases of steam-turbine drives that the limiting conditions which form the basis of the present paper were put into mathematical form so as to enable their use in gear design.

**A.S.M.E. Annual Meeting, December 1-4, 1924, New York, N. Y.**

**The technical program for the coming Annual Meeting gives promise of a great wealth of important material, with practically all Divisions of the Society preparing to present papers. As usual the National Exposition of Power and Mechanical Engineering will be held in the Grand Central Palace parallel to the Meeting. The social events being arranged will interesting innovations.**

**On page 722 of this issue will be found the details of the methods of distributing the Annual Meeting papers while on page 726 will be found abstracts of those papers to be published in pamphlet form.**

**The current issues of the A.S.M.E. News are carrying the details of the Meeting.**

MECHANICAL ENGINEERING

Volume 46 November, 1924 No. 11

# The Zoelly Turbine-Driven Locomotive

## Particulars of a 1000-hp. Experimental Condensing Locomotive Provided with a Recooler for the Condensing Water—Theoretical Considerations—Fuel Economy as Compared with Ordinary Locomotive

By H. ZOELLY,[1] ZURICH, SWITZERLAND

THE steam locomotive as we know it today is the result of a century of development, but in spite of present-day simplicity and reliability the attendant drawbacks are still lack of efficiency and insufficient capacity. For any one familiar with the efficiencies of modern steam plants it is positively pitiful to see hundreds of thousands of locomotives running about with maximum overall efficiencies of 7 to 9 per cent—very often as low as 5 to 6 per cent—whereas in up-to-date stationary steam plants efficiencies up to and in excess of 20 per cent are attained with ease.

The inefficiency of the steam locomotive was recognized long ago, however, and in order to improve matters many railway lines were electrified. Large and powerful power stations were built and equipped with high-efficiency turbo-generators, and notwithstanding the incidental transformation of power a very much higher degree of efficiency was obtained than with the ordinary locomotive. At the same time a second advantage was also realized, namely, that the capacity of the electric locomotive could be increased. Notwithstanding the great advantages offered by this solution, however, there are great drawbacks attached thereto, such as enormous costs, complete dependence on a few generating stations, and long transmission lines. The great war put a stop to electrification schemes practically everywhere, while at the same time the cost of coal rose to an extent previously unheard of. Under these circumstances the need of improving the existing locomotive became imperative. The direction in which improvement lay was clearly indicated: it meant following up the development of stationary steam plants. In the average stationary steam plant the steam supplied by the boiler has a total heat of 1350 B.t.u.—steam at 213 lb. and 662 deg. fahr. With a vacuum of 28.55 in., 427 B.t.u. are available for transmission into energy and 923 B.t.u. are lost in the condenser. In the case of an ordinary locomotive working with the same initial steam only 236 B.t.u. could be used, 1114 B.t.u. being lost owing to the locomotive's exhausting into the atmosphere. To improve the locomotive, then, it is necessary to improve the ratio of the number of heat units used to that of those lost in the process, which can evidently be done in two ways. One way would be to introduce condensation, thus giving similar conditions to those prevailing in the case of stationary steam plants. The other, which would of course represent an improvement for both locomotive and stationary plants, is to increase the steam pressures to values much higher than those obtaining today. If it were possible to increase the boiler pressure to 1434 lb. per sq. in. and the total temperature to 750 deg. fahr., the total heat of the steam would be 1329 B.t.u., and if again we expanded down to 28.55 in. the total heat lost would be only 810 B.t.u., while 519 B.t.u. could be utilized. The advantage thus gained would be enormous, for not only would the available heat drop be much larger, but at the same time the total heat of the initial steam would be less. For the present-day locomotive—exhausting into the atmosphere—the available heat drop would be 353 B.t.u., and 976 B.t.u. would be lost. This would constitute a remarkable improvement on what actually takes place, but it would still be nothing compared with the improvement to be obtained by introducing condensation. It scarcely needs to be pointed out that in the case of the steam locomotive high pressures should only be introduced after successful experience in connection with stationary plants, conditions on a locomotive being so extremely difficult and considerations of reliability and safety having to come before everything else.

FIG. 1 ZOELLY 1000-HP. TURBO-LOCOMOTIVE

The greatest improvement to the existing locomotive consists, therefore, in the introduction of condensation. Without going into details, it may be said at once that for the low-pressure part of the condensing locomotive at least, it will be impossible to make use of cylinders, as the dimensions required for utilizing the relatively good vacua would be too great for them to be housed on the locomotive. For this reason it is only natural that the practice which has been followed in the case of modern power plants should be adopted, namely, the introduction of turbine drive combined with condensation. All engineers working on the subject of improving the steam locomotive as we know it today have come independently to the same conclusion. Among those that may be mentioned in this connection are Ramsay and Reid-MacLeod in Great Britain, and Ljungström in Sweden. The means employed in realizing the fundamental principle, however, were entirely different. The following gives a detailed account of the principles applied in the turbine-driven condensing locomotive built by Escher Wyss & Co., Zurich, in conjunction with the Swiss Locomotive Works, Winterthur, both in Switzerland, according to the Zoelly patents.

### GENERAL DESCRIPTION OF THE ZOELLY TURBINE-DRIVEN LOCOMOTIVE

The locomotive in its present form is shown in Figs. 1 and 2.

[1] Chairman of Board of Directors, Escher Wyss & Co.

Contributed by the Railroad Division for presentation at the Annual Meeting of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS, New York, Dec. 1 to 4, 1924. Abridged. All papers are subject to revision.

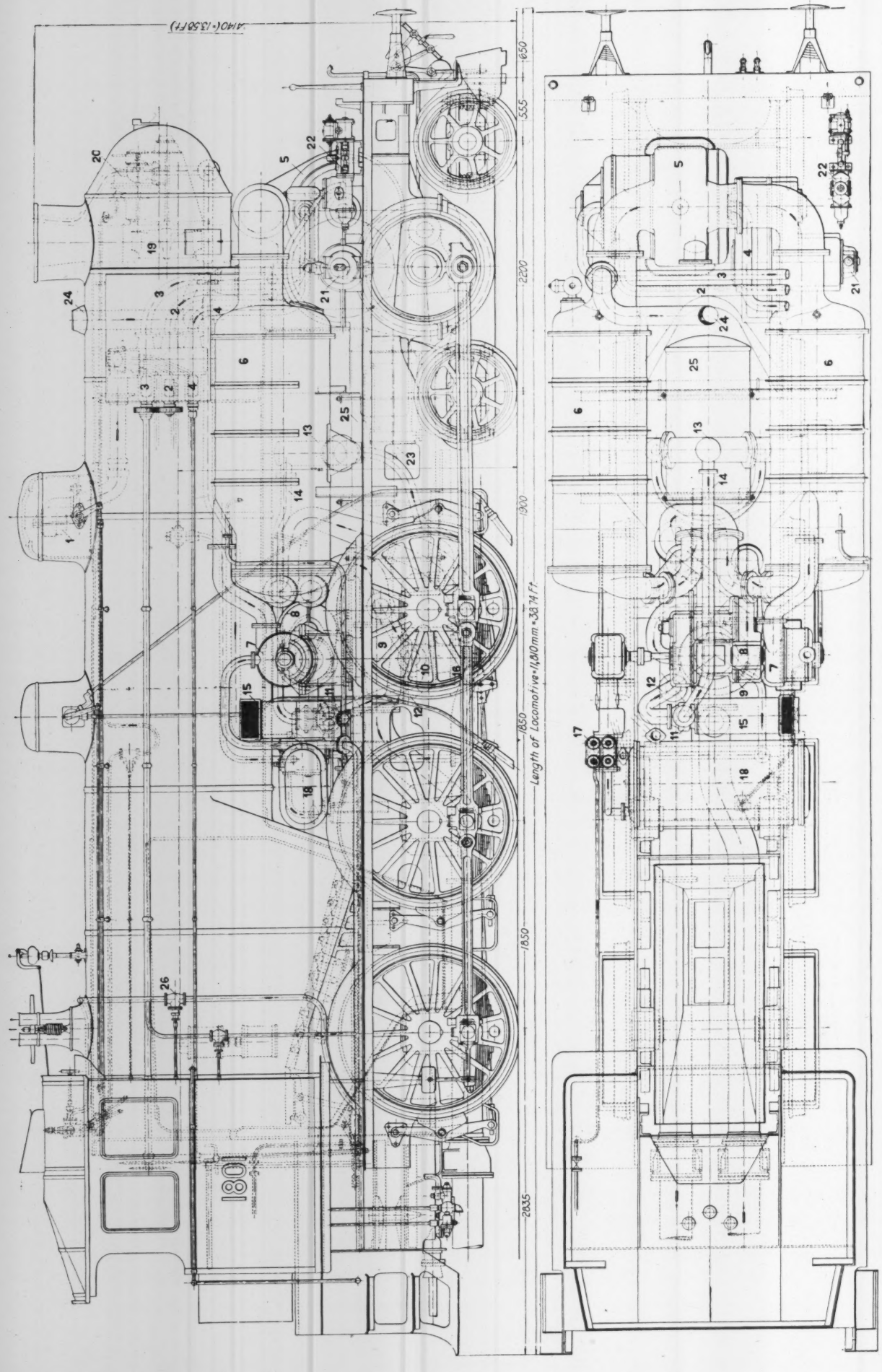


Fig 2 Plan and Elevation of Zoelly 1000-Hp. Turbo-Locomotive Shown in Fig. 1 (Dimensions in Mm.)

1. Main governor valve
2. Steam valve for running ahead—normal
3. Steam valve for running ahead—overload
4. Steam valve for running astern
5. Main turbines for running astern and ahead
6. Condensers
7. Turbine driving condenser auxiliaries
8. Reduction gear
9. Circulating water pump
10. Centrifugal pump—water for air ejector
11. Water-jet air ejector
12. Water delivery pipe from pump 10 to ejector
13. Air collector for air from condensers
14. Air suction pipe from collector to ejector
15. Air separator
16. Condensate pump
17. Feed pump
18. Feedwater heater
19. Furnace fan
20. Turb e driving the furnace fan
21. Oil pump for main turbines
22. Stand-by oil pump
23. Oil cooler
24. Atmospheric exhaust
25. Air storage for automatic air brake
26. Live steam for turbine on recooler.

This experimental machine was constucted by altering a standard piston locomotive (type B 3/4) belonging to the Swiss Federal Railways. Data of the transformed locomotive are as follows:

| | |
|---|---|
| Maximum speed | 47 mi. per hr. |
| Boiler pressure | 200 lb. |
| Superheat | 662 deg. fahr. |
| Heating surface of furnace | 132.5 sq. ft. |
| Total heating surface | 1060.0 sq. ft. |
| Surface of superheater | 380.0 sq. ft. |
| Total heating surface inclusive of superheater | 1450.0 sq. ft. |
| Grate surface | 25 sq. ft. |
| Diameter of driving wheels | 60 in. |
| Rigid wheelbase | 6 ft. |
| Total wheelbase | 25 ft. |
| Total wheelbase, including tender | 58.4 ft. |
| Weight of locomotive, empty | 60 tons |
| Weight in running order | 65 tons |
| Adhesion weight | 45.6 tons |
| Length over buffers, including tender | 68 ft. |
| Extreme height | 13.8 ft. |

The principal change in general appearance as compared with the original locomotive was the replacement of the cylinders by the turbine.

*Boiler.* The boiler in use is the unchanged one of the original locomotive, provided with a Schmidt superheater. A turbine-driven fan has been arranged in the front part of the smokebox as a substitute for the draft produced by the exhaust steam in the case of the ordinary locomotive.

*Main Turbines.* The new locomotive has been designed for the same performance as the old one, so that the turbine for running ahead is designed to give 1000 hp. at the crankpin. The turbine itself is a 6-stage impulse Zoelly turbine. The astern turbine consists of a simple compound wheel and is erected in the same casing as the ahead turbine (see Fig. 3). The turbine rotor, comprising both ahead and astern wheels, is made out of a solid block, the blades being inserted in slots in the wheel rims in accordance with approved practice. The turbine drives through double-reduction gear (1st reduction, 1:7; 2nd, 1:4.1) a jackshaft carrying the crank and crankpins, the drive to the wheels being obtained in the usual way.

The turbine casing with reduction gear, intermediate shaft, jackshaft, and all bearings are mounted on a one-piece steel casing which is riveted to the locomotive frames. The turbine is placed in front of the boiler, its axis being parallel to the locomotive axles.

Referring to Fig. 2, steam admission to the ahead or astern turbines is controlled by means of valves 2 and 3 for ahead and 4 for astern, which are operated by hand from the driver's cab. For running ahead two groups of nozzles have been provided in the first guide wheel, one allowing the passage of about 11,000 lb. of steam when fully open, and the other 4400 lb. According to load, one or the other of the valves—or both—is fully open. Intermediate quantities are obtained by throttling with the main governing valve 1. For running astern only one valve has been provided, allowing a total of 15,500 lb. of steam to pass. Smaller quantities are likewise obtained by throttling down with valve 1. The efficiency of the astern turbine is lower than that of the ahead turbine, but a locomotive must for other reasons run ahead as a rule, astern running only being provided for switching, maneuvering, or in case of emergency.

The maximum traveling speed of the locomotive is 47 miles per hour and is limited by the type of locomotive, the driving wheels having a diameter of only 60 in. At this speed the turbine makes 7500 r.p.m. The turbine speed is of course proportional to the traveling speed. When running ahead the astern wheel rotates in the vacuum, as is generally the case in marine propulsion. The wheel friction for the simple astern turbine is small, and the losses are therefore insignificant.

*Condensers.* From the exhaust end of the turbine—common to both the ahead and the astern turbine—the steam passes in about equal quantities to condensers placed longitudinally on each side of the boiler. These condensers are water cooled and are of the surface type.

*Auxiliaries.* Referring to Fig. 2, all auxiliaries of the condensing plant are driven by one small turbine 7 revolving at 9000 r.p.m. This turbine, through a reduction gear which lowers its speed to 1200 r.p.m. and bevel gearing, drives a vertical shaft carrying the circulating-water, air, and condensate pumps. The circulating pump takes water from the tender, forcing it through the condensers and back again to the recooler. The air pump discharges at a pressure of about 75 lb. to the water-jet air ejector 11, Fig. 2, which air ejector communicates with the two condensers. In the air separator 15 the air can escape into the atmosphere, the water returning to the suction side of the circulating pump.

The condensate from the condensers is led to the condensate pump, which latter discharges at about atmospheric pressure into the feed pump 17 placed on the side platform of the locomotive. This is a reciprocating pump running at 59 r.p.m. only, but driven through a second reduction from the auxiliary turbine. The feed pump discharges direct through a heater into the boiler.

The turbine is a three-stage Zoelly impulse turbine and receives steam at 11 lb. gage pressure, being connected to the condenser. The steam used in the turbine is exhaust steam from a back-pressure turbine driving the ventilator of the recooler.

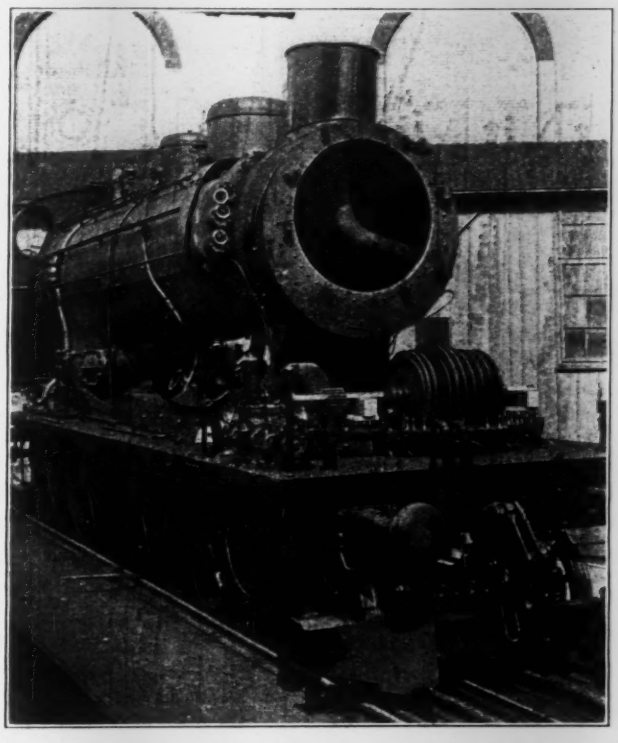

FIG. 3 ROTOR AND FIRST REDUCTION GEAR OF ZOELLY LOCOMOTIVE

*Furnace Fan.* For producing the furnace draft special means have to be provided, exhaust steam being no longer available. In its earliest form the locomotive was equipped with a forced draft producing pressure under the grate. After a long series of tests, however, it was found necessary to change over to the suction principle and to adopt the solution shown in Fig. 2.

The fan is of the centrifugal type, provided with a spiral casing, and is capable of producing a depression in the smokebox of 8.2 in. of water. A maximum of 280 cu. ft. of flue gases can be discharged per second while running at 1500 r.p.m. The fan is driven by a small turbine through a gear with a transmission ratio of 1:6. The turbine receives live steam and exhausts with a back pressure of about 7 lb. gage to the feedwater heater. The admission of steam to the turbine is regulated by a valve operated from the engineer's cab. The connection of the heater with the exhaust steam of the turbine driving the furnace fan is very useful, as there is a certain ratio between the quantity of feedwater and the exhaust steam from that turbine. If for some reason or other there should be no water in the heater and the steam could therefore not condense, a safety valve opens a connection from the heater to the condenser. The condensate of the heating steam always escapes directly to the condenser and, together with the condensate of the main circuit, goes to the boiler.

This design adopted for the experimental machine lacks the advantage of the usual draft producer, i.e., the proportioning of the

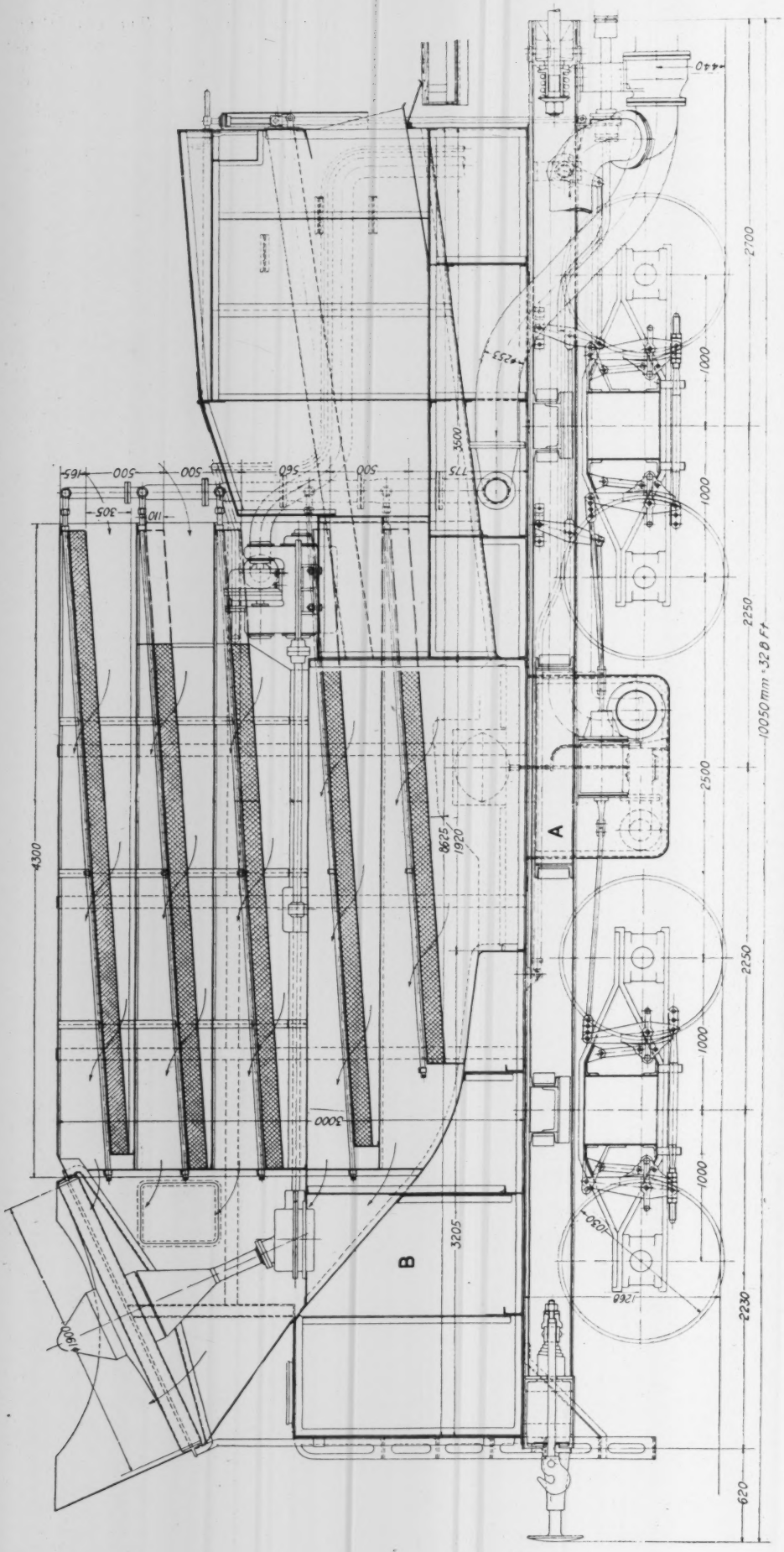


FIG. 4 RECOOLER FOR ZOELLY 1000-HP. TURBO-LOCOMOTIVE (DIMENSIONS IN MM.)

draft to the quantity of steam required in the main turbine. This, however, can be realized on condensing locomotives by bleeding steam from the main turbine.

*Air Pump for Automatic Air Brake.* On the experimental machine the usual Westinghouse air pump—exhausting to atmosphere—is used. The design is not very appropriate, however, as the exhaust steam is lost, and in any case could not very well be used on account of the oil it contains. In future designs the natural course will be to employ a rotary pump, which can be driven by the same auxiliary turbine that drives the condensing auxiliaries, thus returning all steam to the boiler.

*Boiler Feedwater.* In principle the condensing locomotive does not need any additional water for boiler feeding, as the water in the boiler is working on a closed circuit. Practically, however, it is impossible to eliminate leakage losses, loss through the steam whistle, and last but not least, loss through train heating. In order to obtain the full advantage from the condensing locomotive it is essential that none other than clean, soft water gets to the boiler. This can be done in two different ways. It is possible to have a special tank for boiler feedwater on the tender, the feeding being effected by injector as in the case of the ordinary locomotive, or water from the cooling-water tank may be used, which water has to be cleaned before being sent to the boiler. The Krupp Company, of Essen, which holds a Zoelly license, cleans the make-up water by sending it to a small evaporator. The feedwater evaporated in the evaporator escapes into the condenser, where it condenses and is sent to the boiler along with the condensate of the main circuit. Instead of leading the steam from the evaporator direct to the condenser, it is also possible to send it to a low-pressure turbine, or to a certain stage of the machine, thus doing useful work.

In the experimental machine neither of these two solutions has been resorted to. For boiler feeding the cooling water direct is used, being fed by the steam injector if necessary.

*Lubrication.* Each turbine has its own lubricating system, comprising an oil tank and a geared pump driven from a shaft of the reduction gear. Gears and bearings are under forced lubrication, the oil for the main turbines passing through an oil cooler connected with the cooling-water circuit of the condensers.

*Recooler.* The most vital part of the condensing locomotive working with water as refrigerating medium in the condenser is the recooler. All the heat units taken from the steam in the condenser go to the cooling water, which of course has to be recooled in order to be able to be used in a cyclic process.

The recooler is a separate vehicle taking the place of the usual tender and providing room for coal and eventually make-up

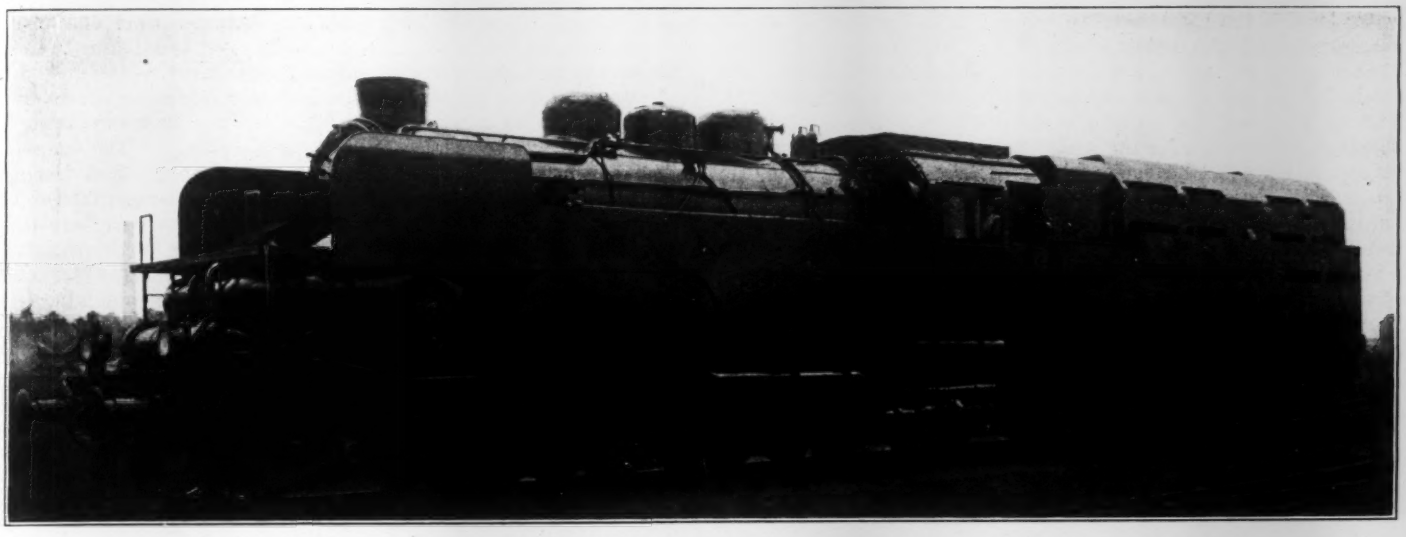

FIG. 5 2000-HP. ZOELLY TURBO-LOCOMOTIVE BUILT BY KRUPP

water for boiler feeding. It works on the vaporization principle, air being brought into very intimate contact with the water to be cooled, and being thus heated and saturated with water vapor. The heat necessary for evaporation is withdrawn from the water to be cooled. The heat to be absorbed in such manner is enormous, amounting in the case of the 1000-hp. experimental machine to about 5,760,000 B.t.u. per hour, and increasing in proportion for larger machines.

The recooler comprises a certain number of elements working in parallel as regards water and air. Each element consists of a wrought-iron channel of practically rectangular cross-section. In the longitudinal direction this channel is divided diagonally into two halves by means of perforated trays containing Raschig rings, i.e., small tubes of about equal length and diameter. The water to be cooled is led to these trays by tubes acting as sprayers, the air passing in counterflow. In the experimental cooler shown in Fig. 4 the cooling elements are so disposed that the natural current of air produced by the traveling train can enter the cooling element direct. The fan provided produces a sufficient current of air when the train is stationary or traveling at a very slow speed, and also augments the normal current when traveling at ordinary speeds. It is driven through a gear by a small back-pressure turbine, as mentioned before. Admission of steam to the turbine is controlled by means of a valve from the engineer's cab.

The cooled water flows from the cooling element back into a tank, whence it is again drawn into the circulating pump. As a certain amount of water is evaporated in the process it becomes necessary to add a corresponding amount in order to keep the circulating quantity constant. For this purpose there is provided the large storage tank *B* on the tender, which communicates with the suction tank *A* by means of floaters.

The water in the tender need not necessarily be so clean and pure, as it does not enter the boiler nor come into contact with such parts as could seriously affect the working of the plant.

Room is provided in the tender for 11 tons of coal and 6 tons of water.

The recooler car and locomotive are coupled in the usual way. The connections of both suction and discharge pipes for the cooling-water system are made by means of sliding ball-and-socket joints. On removing the coupling bolts these allow of the tender's being taken away from the locomotive. Connections for live steam and for exhaust steam from the turbine driving the fan of the recooler are of very small dimensions only and are made by flexible pipes.

Before dealing with the theory of the most important parts of the turbine-driven locomotive and its superiority over the ordinary locomotive, a brief sketch will be given of the 2000-hp. locomotive built at the Krupp works under license to use the Zoelly patents.

## 2000-HP. ZOELLY TURBO-LOCOMOTIVE BUILT BY KRUPP

Fig. 5 shows the Krupp locomotive, which is now completed and will shortly be tested. The general arrangement is absolutely identical with that of the Zoelly experimental machine, the only differences being in sundry details. Having twice the energy to transmit, it was found necessary to locate the reduction gear in about the middle of the driven axle. For this reason the turbine has been divided into separate ahead and astern turbines, placed symmetrically to the locomotive axle outside the longitudinal bars of the frame.

The turbines and gears, including intermediate shaft and jackshaft, are mounted on a one-piece steel casting which can be slid as a whole on to the main bar frames of the locomotive. The latter extend through the casting and carry the buffers.

The condensers are led crosswise to the boiler, under the latter, and are connected in series. The condensing auxiliaries are in principle arranged as on the experimental machine. The furnace fan is mounted just in front of the firebox under the boiler and is connected to the smokebox by a large wrought-iron channel. In this channel is arranged a feedwater heater. The fan exhausts again in a similar channel leading to the stack. The turbines driving the furnace fan as well as the turbine driving the condensing auxiliaries are condensing machines fed with live steam. The turbine driving the fan of the recooler is a back-pressure turbine exhausting into a feedwater heater arranged in series and ahead of the one already mentioned.

The cooling elements are arranged crosswise to the axis of the tender and symmetrically to it. They communicate on one side with the atmosphere and on the other with a central channel leading to a centrifugal fan. A double fan may be used.

## THEORETICAL CONSIDERATIONS

*Recooler.* The recooler will be considered first, as the vacuum obtainable determines the calculation of the turbines and auxiliaries. The better the vacuum, the greater the total heat which can be transferred to useful work. It is known from stationary plants that with surface condensers the vacuum directly depends on the cooling-water temperature. In the case of the condensing locomotive the vacuum therefore depends upon the temperature to which the cooling water can be lowered in the recooler.

The maximum temperature that the air going through the recooler can attain is that of the warm cooling water. If it be assumed that the air at this maximum temperature is fully saturated with water vapor, then the difference in total heat between the air entering and that leaving the recooler is identical with the amount of heat withdrawn from the water. When a turbine is working against different vacua it is possible, assuming definite initial steam conditions and constant turbine efficiency, to calculate the amount of heat which has to be extracted from the steam in the condenser or, what is the same thing, from the cooling water in the recooler. If it be further assumed that the vacuum in the turbine exhaust corresponds to the water-vapor tension of water at a temperature which is 41 deg. fahr. higher than the cooling

water leaving the condenser, we can calculate the amount of air necessary to obtain a certain vacuum for a given quantity of steam. This calculation has been made and in Fig. 6 curve I shows the result for cooling air at 59 deg. fahr. and 70 per cent saturated. The air necessary for cooling is delivered by a fan. Assuming constant surface area for the cooler for all the different vacua, the resistance through the cooler would of course increase with the quantity of air, and the power required for driving the fan would augment exceedingly rapidly with better vacua. These conditions are illustrated in curve II, the horsepower referring to the power necessary for each pound of steam per second. A long series of tests has shown that for full load the vacuum will be somewhere near 90 per cent, which of course would only apply to maximum

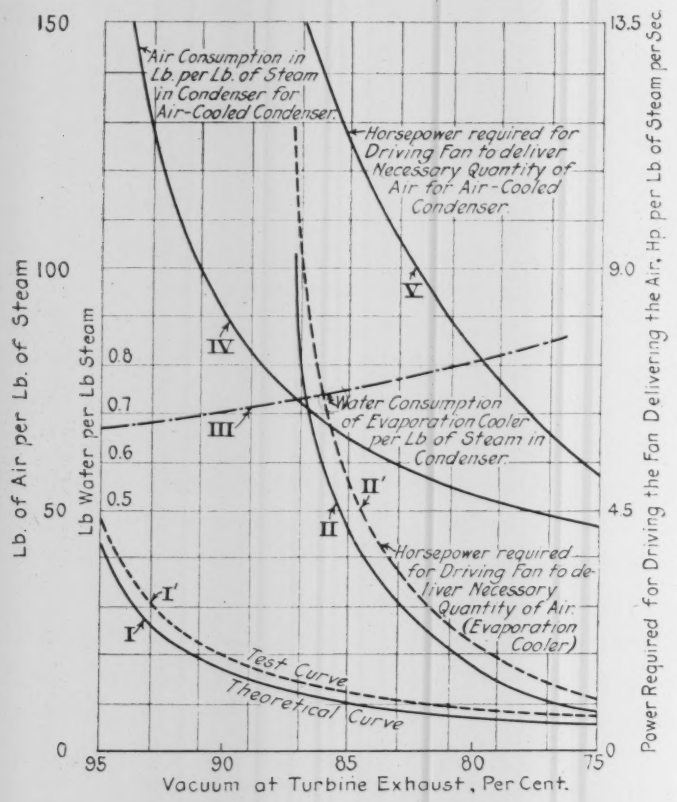
Air Consumption in Lb. per Lb. of Steam in Condenser for Air-Cooled Condenser
Horsepower required for Driving Fan to deliver Necessary Quantity of Air for Air-Cooled Condenser
Water Consumption of Evaporation Cooler per Lb of Steam in Condenser
Horsepower required for Driving Fan to deliver Necessary Quantity of Air (Evaporation Cooler)
Test Curve
Theoretical Curve
Lb. of Air per Lb. of Steam
Lb Water per Lb Steam
Power Required for Driving the Fan Delivering the Air, Hp per Lb of Steam per Sec.
Vacuum at Turbine Exhaust, Per Cent.

FIG. 6 AIR CONSUMPTION FOR EVAPORATION AND SURFACE COOLERS

loads. On partial loads the vacuum would increase on the amount of steam sent to the condenser being less.

Curve III gives the water consumption of the recooler in pounds per pound of steam.

Curves I′ and II′ give test results (air quantity and power absorbed by fan) which show that actual conditions do not differ materially from the theoretical.

To show the efficiency of the Zoelly cooling system as compared with that obtained by employing an air-cooled condenser, curve IV in Fig. 6 gives the necessary quantity of air on the assumption that the air could warm up to 122 deg. fahr., and curve V the power required for the fan per pound of steam per second. The difference as compared with the evaporator cooler is enormous. The Zoelly system of cooling involves the necessity of having a condenser and recooler, but affords important advantages. The coefficient of heat transmission from steam to water in a surface condenser is about 490 B.t.u. per deg. fahr. per sq. ft. per hr., whereas that for air-cooled condensers from steam to air is only a trifle over 8 B.t.u.[1] The surface of the air condenser must therefore be 60 times that of a water-cooled condenser. This makes it possible to have on the Zoelly locomotive both condenser and turbines on the same truck as the boiler, thus overcoming all difficulties involved in having vacuum connections between tender and locomotive. The relatively small vacuum space is easily kept airtight, while the system further allows of the locomotive's being the real driving part, thus adhering to this extent to the old and conventional design.

It was of course impossible to calculate the recooler, and very extensive tests were necessary to clear up the very complicated relationships and to get the data for actual design. The complicated nature of the task of finding the most economical design, i.e., getting the maximum of cooling effect for a given surface with the minimum of weight and driving power, will be seen from the numerous variable factors, viz., material of filling bodies, height, quantity of water, velocity of air, distribution of water, water and air temperatures, etc. As a result of all tests it was found that for average conditions—air 59 deg. fahr., cooling water 122 deg. fahr., corresponding to about 85 per cent vacuum—a cooling effect of 44,260 B.t.u. per sq. ft. an hour is obtained.

There still remain many interesting details in connection with this cooler, but their discussion would lead too far for the present paper, the scope of which is confined to giving a general idea only of the Zoelly turbo-locomotive.

*Main Turbine.* For turbine-driven locomotives superheated steam is used in any case, as this gives the highest efficiency. So long as it is necessary to use the conventional type of boiler it will scarcely be possible to assume other steam conditions than those obtaining for ordinary locomotives, say, 215 lb. pressure and 662 deg. fahr. temperature. The vacuum has been fixed in the

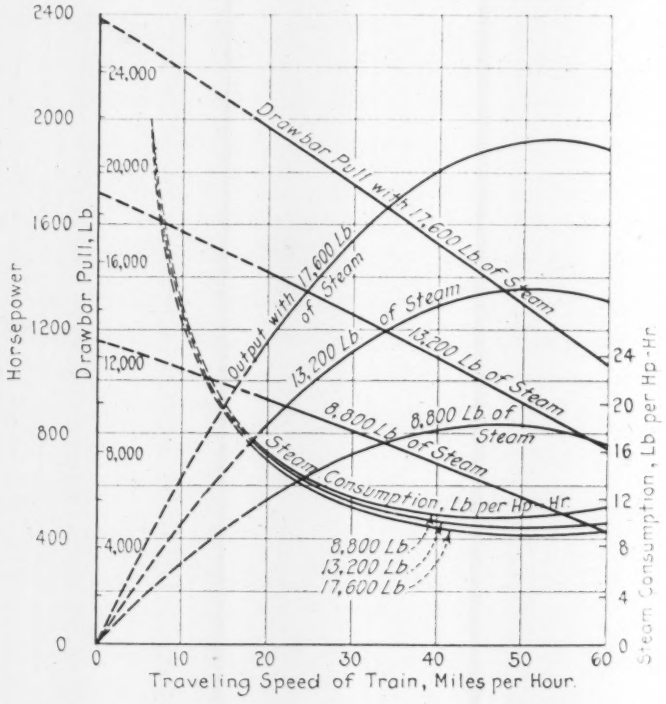
Drawbar Pull with 17,600 Lb of Steam
Output with 17,600 Lb of Steam
13,200 Lb of Steam
8,800 Lb of Steam
Steam Consumption, Lb per Hp-Hr
8,800 Lb
13,200 Lb
17,600 Lb
Horsepower
Drawbar Pull, Lb
Steam Consumption, Lb per Hp-Hr
Traveling Speed of Train, Miles per Hour

FIG. 7 OUTPUT, DRAWBAR PULL, AND STEAM CONSUMPTION OF 2000-HP. ZOELLY TURBO-LOCOMOTIVE FOR DIFFERENT SPEEDS AND STEAM QUANTITIES

preceding paragraphs and the output desired is given, so that as variable factors in the turbine calculation there are the turbine speed, the wheel diameters, and the number of stages. These elements have to be so chosen that the efficiency will be as high as possible. On the locomotive we are limited in the matter of wheel diameters as well as in the number of stages. If these are determined the speed is dictated, any departure from which speed would mean a certain loss in efficiency, which within relatively wide limits is not of the utmost importance. The turbine speed determines the gear ratio and dimensions. For normal conditions the turbine speed ranges from 6000 to 8000 r.p.m., but it also depends upon the maximum speed the train can attain, this in turn being limited by the stresses allowed.

A locomotive turbine has to be calculated not only for one particular speed and one load, as is usually done for stationary plants, but for different quantities of steam flow and for different speeds for each steam flow, in order to obtain full information

[1] R. P. Wagner, *Organ fur die Fortschritte des Eisenbahnwesens*, Jan. 15, 1924, p. 3.

as to the performance of the locomotive. Such curves as calculated for the Krupp locomotive, showing output and drawbar pull for different speeds and steam quantities, are given in Fig. 7. For running astern a sacrifice in efficiency is admitted on account of the extreme limitation of space. The only point to consider is that sufficient initial torque must be available for accelerating the heaviest of trains.

*Auxiliary Turbines.* The limitation in size and weight is very important, but efficiency is also exceedingly important. Small wheel diameters call for multiple stages and the turbine would thus be too long. In order to overcome this difficulty several turbines are connected in series. This arrangement in conjunction with high running speeds (9000 to 10,000 r.p.m.) satisfies the demand for high efficiency combined with small dimensions.

*Feedwater Heating.* The heat contained in the smoke gases has long since been utilized in stationary plants, the economizer being a very well-known device. Several feedwater heaters of this type have been tried for locomotives, but without success. The draft available in ordinary locomotives is not sufficient to overcome the additional resistance of such a heater. In turbine-driven locomotives, however, where the draft has to be produced artificially, this difficulty does not present itself and therefore waste-gas feedwater heaters can be employed to advantage. The gases leave the stack with a temperature of about 750 deg. fahr., and can be cooled down to about 350 deg. fahr., thus giving 95 B.t.u. per lb., corresponding to about 1 lb. of feedwater. If the condensate leaves the condenser at a temperature of 122 deg. fahr., it will therefore be possible to heat the feedwater to 217 deg. fahr. The total heat of the steam in the boiler being 1350 B.t.u., the saving effected by heating the feedwater thus amounts to 7.5 per cent. It is a matter of practice to see how much of this heat can actually be utilized.

There are other possibilities of heating the feedwater with steam, which latter can be exhaust steam from auxiliary turbines or steam bled from the main or an auxiliary turbine. Steam being exhausted at atmospheric pressure from a back-pressure turbine, as in the case of the turbine driving the furnace fan of the experimental turbine, has a total heat of about 1150 B.t.u. Were this steam used in a low-pressure turbine working to the condenser, probably 72 B.t.u. could be transferred to useful work, but the steam would enter into the condenser with 1080 B.t.u., of which 990 B.t.u. would go to the cooling water and thus be lost. Using the exhaust steam in the heater, all the 1150 B.t.u. can be utilized. Under normal steam conditions the feedwater can be heated up to about 302 deg. fahr. The condensate having a temperature of 122 deg. fahr., we can therefore use 180 B.t.u. per lb. of feedwater. The maximum quantity of exhaust steam which can be utilized is accordingly one-fifth of the quantity of condensate. To heat up to this high pressure, of course, steam at a pressure higher than that of the atmosphere has to be used.

The greatest locomotive economy would be effected by preheating the combustion air with smoke gases and the feedwater with steam in a series of feedwater heaters, all heated by steam of different pressures bled from the main turbine or by exhaust steam at suitable pressure. Steam bled from the main turbine or steam from the furnace-fan turbine will always be the most convenient, as the proportion of steam to feedwater is adjusted automatically.

## STEAM AND COAL CONSUMPTION

The steam consumption of a turbine working condensing is very much less than that of a reciprocating engine working non-condensing. The condensing machines, however, require a number of auxiliaries also consuming steam and thus lowering the overall efficiency to a certain extent. In calculating the coal consumption it is necessary to consider carefully what preheating system shall be employed, as the coal consumption depends to a very great extent upon this factor.

Table 1 gives the results of such a calculation for the 2000-hp. Zoelly turbo-locomotive built by Krupp and described above, as well as for piston locomotives.[1] The steam and coal economies of the turbo-locomotive are calculated for three different methods of feedwater preheating and also without preheating, and the beneficial effect of employing a suitable heating system is evident

## TEST RESULTS—FUTURE PROSPECTS

The Krupp locomotive has only just been completed, so that test results will not be available before a certain lapse of time. Tests have been carried out with the experimental machine, but only over a very short line (35 miles) with very variable grades, so that there was no possibility of making coal-consumption tests. All that could be determined was the steam consumption, which may be compared with that of a corresponding piston locomotive making the same journey with the same load. Considering the preheating employed on the turbo-locomotive and the fact that for the auxiliaries only saturated steam was used, the consumption of heat in the turbo-locomotive for the round trip was only 4,230,000 B.t.u., whereas that of an ordinary locomotive—same train and identical conditions—was 5,904,000 B.t.u. (calculated from the difference in water level in the tender before and after the run and the initial steam conditions.) The tests were repeated four times, the average being given. The water consumption of the recooler was 11,880 lb. Of this quite a large amount represented mechanical loss through leakage.

In spite of the various parts comprised in a turbo-locomotive, its operation is simple and less attention on the part of the personnel is called for than is the case with the ordinary locomotive. Before starting the engineer speeds up the turbine driving the fan on the recooler, which operation consists solely in opening the corresponding valve. As this turbine is connected in series with the auxiliary turbine driving the condensing auxiliaries, the condensing plant is started automatically. After the working vacuum has been attained, which takes several minutes, the steam-admission valve to the main turbine is opened wide, and when it is desired to start the train it is only necessary to open the governor valve. On the experimental machine it is also necessary to have the furnace fan started, which, however, only requires a valve to be opened. The opening of this valve is governed in such a way

[1] From *Kruppsche Monatshefte*, Essen, Germany, 1924, p. 24.

TABLE 1 COMPARATIVE STEAM AND COAL CONSUMPTIONS OF DIFFERENT LOCOMOTIVE TYPES

| | Piston-Locomotives, Non-Condensing | | | | Piston Locomotives, Condensing | | Turbo-Locomotive | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | Superheated-Steam Simple 2-Cyl. | | Superheated-steam 4-Cyl Compound | | Superheated steam Simple, | Compound | ——With Feedwater Heating—— | | | |
| | Without preheating | With preheating | Without preheating | With preheating | Without preheating | With preheating | Without preheating | With steam | With smoke gases | Preheating feedwater and combustion air |
| Boiler pressure, lb. per sq. in. abs | 213 | 213 | 213 | 213 | 213 | 213 | 213 | 213 | 213 | 213 |
| Pressure at steam chest, lb. per sq. in. abs. | 199 | 199 | 199 | 199 | 199 | 199 | 199 | 199 | 199 | 199 |
| Temperature in steam chest, deg. fahr. | 572 | 572 | 572 | 572 | 572 | 572 | 662 | 662 | 662 | 662 |
| Exhaust pressure, lb. per sq. in. abs. | 14.2 | 14.2 | 14.2 | 14.2 | 2.13 | 2.13 | 2.13 | 2.13 | 2.13 | 2.13 |
| Steam per i.hp-hr. (including all auxiliaries), lb. | 16.97–16.28 | 16.97–16.28 | 16.5–15.61 | 16.5–15.61 | 13.83 | 11.55 | 10.59 | 11.08 | 11.08 | 11.08 |
| Saving in steam consumption compared to simple locomotive working with superheated steam, non-condensing, lb. | 0 | 0 | 0.47–0.67 | 0.47–0.67 | 3.14–2.45 | 5.42–4.73 | 6.38–5.69 | 5.89–5.20 | 5.89–5.20 | 5.89–5.20 |
| Do., per cent | 0 | 0 | 2.76–4.1 | 2.76–4.1 | 18.5–15.1 | 31.9–29.0 | 37.6–35.0 | 34.7–31.9 | 34.7–31.9 | 34.7–31.9 |
| Total heat required in boiler per hp-hr., B.t.u. | 44,860–41,680 | 33,830–32,470 | 36,850–34,880 | 32,910–31,140 | 30,410 | 25,510 | 20,120 | 19,510 | 17,980 | 16,410 |
| Coal consumption (heating value of 11,700 B.t.u. assumed), lb. per hp-hr. | 3.23–3.10 | 2.88–2.77 | 3.14–2.97 | 2.81–2.52 | 2.60 | 2.18 | 1.72 | 1.67 | 1.54 | 1.40 |
| Saving in coal compared to a simple locomotive working with superheated steam and feedwater heating, lb. | —0.35 to —0.33 | 0 | —0.26 to —0.20 | 0.07–0.25 | 0.28–0.17 | 0.70–0.59 | 1.16–1.05 | 1.21–1.10 | 1.34–1.23 | 1.48–1.37 |
| Do., per cent | —12.4 to —12.1 | 0 | —9.3 to —8.9 | 2.43–8.9 | 9.3–5.65 | 24.1–21.0 | 40.0–38.0 | 41.8–39.5 | 46.5–44.4 | 51.1–49.2 |

that the boiler pressure is kept constant. Boiler feeding is no longer necessary. To keep the locomotive going at a certain speed it is no longer necessary to vary the degree of admission; all that is required is simply to throttle the steam admission to the required extent. If for any reason the output is insufficient with the governor valve wide open, then the overload valve allows an increase.

All that is required to stop the locomotive is to close the governor valve. The auxiliaries are not to be shut down unless for a prolonged stop.

To change over from running ahead to running astern, after first having stopped the machine, the valve for running ahead is closed and that for running astern is opened; after which all that is necessary is to open the governing valve.

Starting up takes place very smoothly, and the fears expressed by many engineers that a turbine-driven locomotive would not have sufficient initial torque, are absolutely unfounded.

The running of the turbine-driven machine is also very smooth compared with that of a piston locomotive, this being on account of the absence of reciprocating masses. The drawbar pull also is constant, as is shown by the smooth line which is obtained in a diagram taken with the dynamometer of the Swiss Federal Railways.

The experimental locomotive described has a capacity of 1000 hp. and that built by Krupp one of 2000 hp., but this is by no means the limit of possibilities. Studies and complete designs have been made for locomotives up to 5000 hp., neither locomotive nor recooler presenting any particular difficulty.

Today a further development of the turbine-driven locomotive is impending, namely, the employment of very high pressures, their practical utilization being only a matter of finding a suitable boiler. As far as the turbine is concerned, the designs are all ready.

The advantages of the turbo-locomotive are manifold: high economy in fuel, amounting to very nearly 50 per cent; low water consumption, the water not necessarily being good water for the purpose; boiler kept clean; very smooth running on account of the absence of reciprocating masses, etc. There can be no doubt that the turbine-driven locomotive will very soon enter into successful competition not only with the old piston locomotive but also with schemes of electrification which are expected to result in greater efficiency in the utilization of coal. Its greater economy gives increased capacity with the same boiler, thus opening a new era of steam-locomotive traction.

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# The Measurement of Management

IN A PAPER on this subject which was presented simultaneously at the Chattanooga Regional Meeting of the A.S.M.E. and at a joint meeting of interested societies in New York on October 23, 1923, the author, Prof. J. W. Roe, outlined the problem and suggested a method for comparing managerial performance in different plants—or of variations, thereof in a single plant—that he believed would be trustworthy, practically useful, and as free as possible from the element of personal opinion. One portion of Professor Roe's paper—that dealing with the measurement of the cost-accounting function—was made the topic of a discussion at the management session of the A.S.M.E. Spring Meeting in Cleveland last May, extracts from which were published in MECHANICAL ENGINEERING for August, page 477; while another—that on measurement of the quality of product—formed the subject of a paper presented by G. S. Radford at the same session and published in the September issue, page 546. Professor Roe's closure to these discussions follows.

### PROFESSOR ROE'S CLOSURE

Discussion at this session has centered on two of the ratios suggested in the paper: the ones relating to quality and cost accounting.

In connection with the quality ratio Mr. Radford suggests that the customer's inspection should be excluded from the numerator. What was in mind in the original paper was not the detailed results of the customers' inspection, which would seldom be obtainable, but their rejections due to failure in quality. Mr. Radford himself unconsciously gives the reason why the ratio should include the customers' rejections. If the numerator be confined to the units which pass the producer's inspection, it would, as he points out, cover only "the number of articles which the factory inspection considers commercially salable." The quality standards might be set so low as to permit all of the goods to pass plant inspection but the customers might reject a large proportion as unusable, certainly an unsound condition despite the 100 per cent showing. If the ratio considers also the customer's rejections, this condition would be safeguarded against. The ratio as proposed would tend to bring the plant inspection to that standard which would give the highest rating, i.e., allow the greatest number of units by, consistent with few or no customer's rejections. In view of the fact that a customer's rejection is more serious than one within the plant, this portion of the numerator might be given an extra weight, say, of 2, 5, or 10, based on the judgment of those setting up the ratio.

The purpose of the ratio is not only to measure the quality of production, but to obtain a rating of the *management* with respect to both the correctness of the inspection standards set and also the production performance based on these standards. In the author's judgment both of these elements should be taken into account.

With regard to cost-accounting ratio, the discussion is interesting and suggestive. Mr. Lamb is of the opinion that an evaluation of the cost-accounting and records function is unpractical, "even though formulas could be worked out for that purpose." With this the author cannot agree, as it seems to him quite as desirable that we should know the effectiveness of these important functions as that of any other department. Whether or not it is possible to so evaluate them is a question; the desirability of it, however, seems clear. Mr. Harrison argues for the value and necessity of a good cost system, but does not indicate whether he believes its effectiveness can be measured in any way. Mr. Scanlon believes that the value of a cost system can be measured, but does not indicate how. Mr. Allen feels that it could be reduced to a formula, but that such a formula would be largely theoretical. This would depend upon the formula. If it were confined to measurable elements and did not attempt to be all-inclusive, it might be both practical and useful. Mr. Cutter thinks a measure of efficiency and value can be determined with regard to the accuracy and promptness of information, at least in some cases.

Mr. Cartmell indicates a way of rating cost data as to accuracy and promptness, and adds a further element which it is desirable to follow if possible, namely, the cost of securing this information. His discussion is constructive and is suggestive.

The discussion rather confirms the suggestion in the original paper on the measurement of this function, namely, that we set up an ideal and then rate the performance of the cost department as well as it can be done against this ideal. The first step ought not to be impossible. Good cost accountants are keen-minded and intelligent. It ought to be possible for them to take stock of their own work and to frame some statement, by elements, of what they are actually trying to do and what would constitute an ideal performance of their function, beyond which they could not hope to go. Then let them devise some systematic way of comparing results with this ideal. Mr. Cartmell's discussion would indicate that some elements at least might be measured in a rough way. Other elements cannot be. For these, the practical thing would be to compare the performance with the ideal on the basis of judgment. This should be done at regular periods, upon the same basis, and by the same person. While this is not measurement it is an improvement over haphazard snap judgments.

We must not forget that this function, like all the other functions of management, is actually being judged every day. What the author argues for in the paper is that orderliness and method should be introduced into this judgment, but not beyond the point of good judgment and good sense.

# Die Casting

## Types of Die Casting Machines—Representative Alloys for Use in Die Casting—Die Construction—Casting Problems—Advantages Gained by Use of Inserts

By NORRIS CLEMENTS BARNARD,[1] WORCESTER, MASS.

DIE casting, according to the modern conception, is the art of forcing molten metal under pressure into a mold or die that remains intact for operations after the casting has been removed. The elements necessary to produce successful die castings may be considered from three angles: machines, alloys, and dies.

The machine used must be capable of mounting the die and handling the metal used in the castings. These functions must be uninterrupted. The metal employed must be one that will melt successfully in iron receptacles and produce sound and enduring castings. The construction of the die must be such that it will permit the formation of the casting as designed. The proper execution of these three requirements assures the production of successful castings.

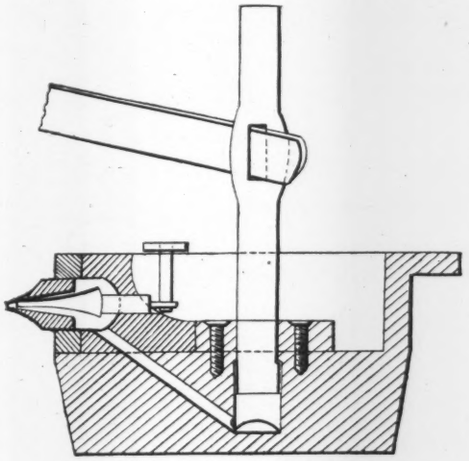


FIG. 1 TYPE-CASTING MACHINE OF 1849

In order to add special local qualities to certain parts of otherwise successful castings, pieces of material possessing these properties are cast into the object, forming an integral part of it. The use of these inserts broadens the field of die casting immensely.

In this paper the author will treat briefly each of the elements suggested above: namely, casting machines, alloys, dies, castings, and inserts.

### DIE-CASTING MACHINES

The first casting machines manufactured type and bullets. The early demand for large quantities of type stimulated the development of such machines a quarter of a century before they came into use for general casting purposes. Type casting is in reality a specialized branch of the die-casting industry.

Fig. 1 shows a type-casting machine patented in 1849. This machine, which is of the plunger type, pumps metal into the mold, and is very simple but very effective. The first die castings were produced on machines of this type, which had in addition to the equipment shown in Fig. 1 a means of mounting and operating the dies.

Some of the machines that are in daily operation at the present time embody the same principle. Fig. 2 gives a cross-section through a modern casting machine of this type. A comparison of Figs. 1 and 2 will show that the two machines are essentially alike, both having the same pumping action to move the metal.

This machine injects the metal into the die between its halves, which constitutes a very simple and direct method of introducing it. The pressure on the molten metal must remain until the metal in the die partly solidifies, so that it will not flow back into the casting chamber on the removal of the pressure.

Fig. 3 shows a section of another machine of this type. In this case the metal passes up from the casting cylinder *b* through the water-cooled gate former 1 into the die 2. As soon as the metal comes to rest after its rush into the die, this cool gate former solidifies it so that it will not flow back into the casting cylinder. This rapid cooling permits the die to operate at a very fast rate, which means lower production costs. The die frame of this machine tilts from a vertical position to that shown in the figure, to remove the gate former from the casting chamber. In the great majority of cases the white-metal dies used on these plunger machines are rather small and light. For this reason the tilting of the die frame does not constitute a serious objection. However, if a simpler motion were substituted, the machine would operate faster and more easily.

The plunger-type machines as illustrated are very effective when using metals having low fusing temperatures, such as lead, tin, and zinc alloys. In the early stages of the industry only lead and tin were employed as in type casting. A short time afterward zinc alloys came into use, and in about 1914 aluminum found its way into the art.

The plunger machines, however, could not successfully handle aluminum with its higher melting point. The metal solidified between the plunger and the cylinder, causing them to stick together. This necessitated the repeated cleaning of these parts, so that casting costs rose to a prohibitive figure.

To eliminate such troubles the overhead machine came into use. As Fig. 4 illustrates, the melting pot and its accessories stand above the die. In the machine shown at (*a*) a plunger *m* rises above the inlet *g*, permitting the metal to flow down into the die cavity.

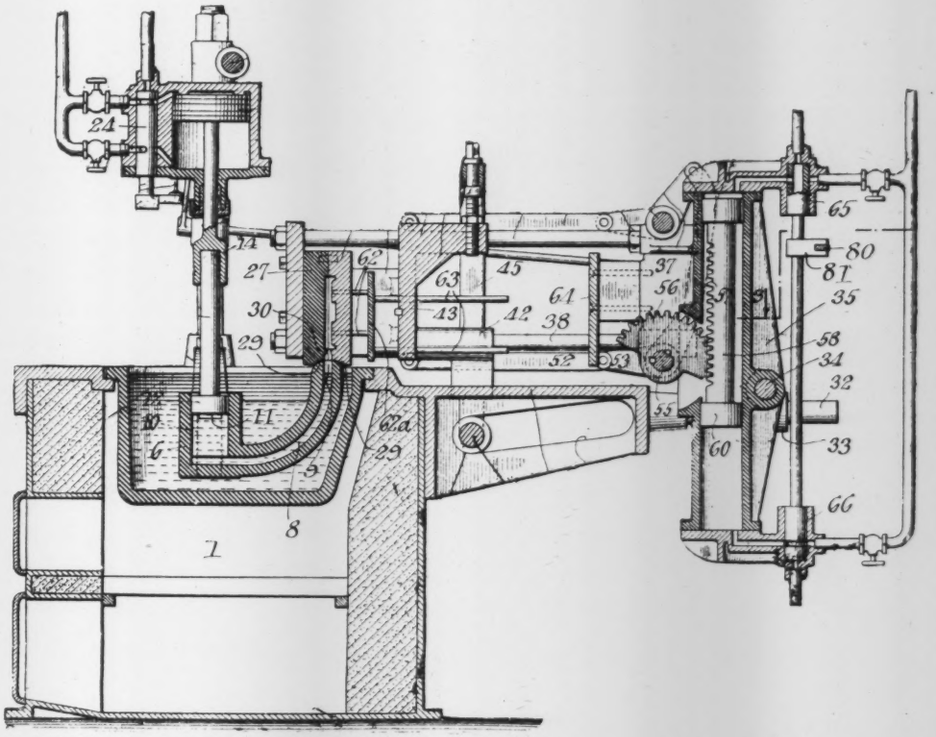



FIG. 2 A MODERN PLUNGER DIE-CASTING MACHINE

[1] Engineer, Boston Pressed Metal, Co., Jun. A.S.M.E.

Presented in competion for the A.S.M.E. Junior Prize, 1924.

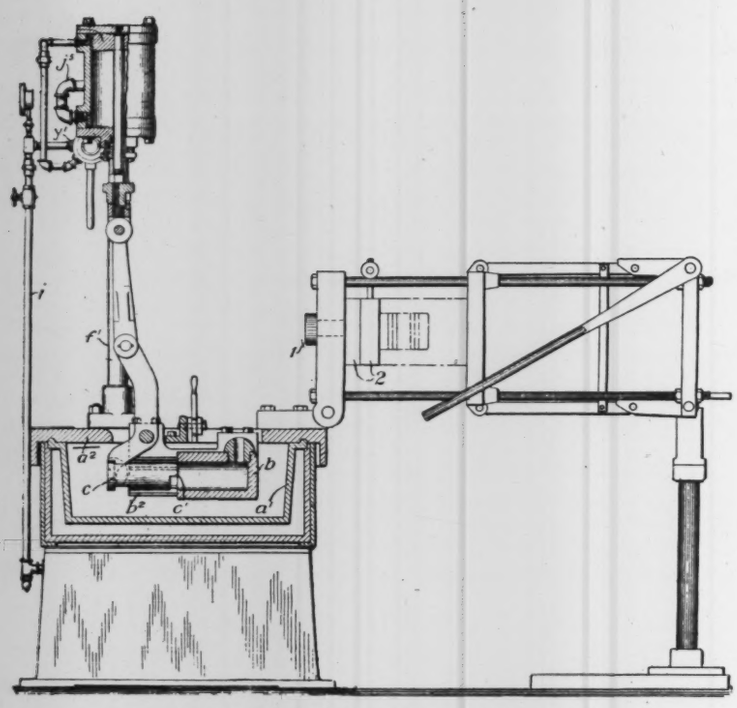


FIG. 3 ANOTHER MODERN PLUNGER MACHINE

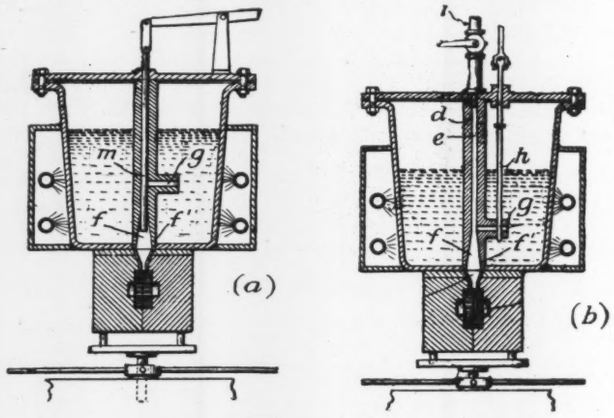


FIG. 4 OVERHEAD MACHINES

When the die fills, the downward motion of the plunger cuts off the flow of the metal at the inlet, and at the same time increases the pressure within the die.

In another construction, shown at (*b*), the valve *g* permits the metal to flow into the die by gravity. Air pressure entering through the pipe *i* after the valve *g* closes, creates the pressure that properly fills the mold.

This type of machine relies upon the direct application of air pressure to the molten metal to obtain a sharply defined casting. Early writers on the subject condemned the use of air in this connection because of its great affinity for liquid metal. However, other troubles developed which proved to be more objectionable. The valve *g* and the plunger *m* of these machines became stuck just as the old plungers did, when using metals of high melting points. Also, it was necessary to lift all fresh charges of metal into the high melting pot. If the valve or the pot became defective, the entire charge of metal flowed down upon the floor and now and then upon an operator in his position under the machine.

In order to eliminate the troubles inherent in all moving parts that are submerged in the molten metal, machines of the type illustrated in Fig. 5 came into use. This machine has no plunger, valves, or equivalent moving parts. Air pressure entering the charging chamber 3 forces the metal into the die. After a casting operation, the charging chamber dips into the melting pot 2 to obtain a fresh charge of metal through its nozzle.

Early predictions that this type of machine would never prove satisfactory, due to the corrosive action of the air upon the molten metal, have not been substantiated. The absence of parts rubbing together while submerged in the metal has insured its popularity. The plunger, as used in the older type of machine, exerts more pressure upon the metal, but air proves more practical for metals with high melting points. This machine is of the same general design as the plunger machine of Fig. 2. The good and the bad features are largely common to both.

Dies mounted on these machines may have cores that pull in five directions: to the right, to three sides of the die—upward, away from the observer, and toward the observer—and short pulls to the left. Due to the close proximity of the melting pot no cores may extend downward. This places a limitation on dies for complicated castings or multiple-impression dies.

In the first firing of these machines the dies, in their position directly over the melting pot and furnace, heat as the metal melts. This is an advantage, in that a warm die produces better castings than a very cold one. The metal will flow more readily, permitting the die to fill more evenly. On the other hand, the work of operating a die-casting machine is very hot at best. In a case such as this, where the operator must work right over the molten metal, the job will prove to be an exceedingly warm one.

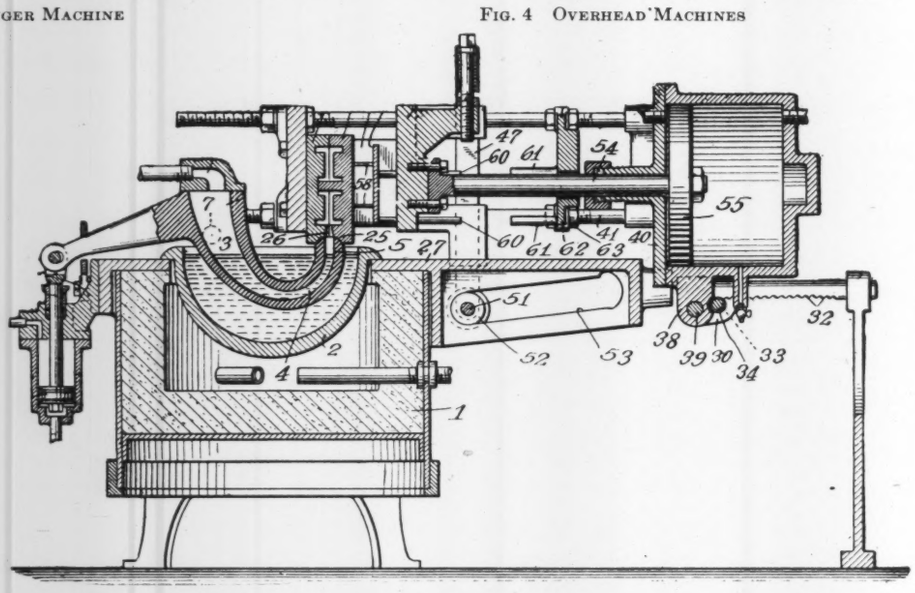


FIG. 5 A MODERN AIR MACHINE

These two machines offer a very direct path through which the metal flows while entering the die cavity. This direct flow tends toward sharply defined castings. The pressure used to force the metal and the inertia of the latter serve to throw the metal into the remote portions of the impression, resulting in clean-cut castings.

Each time the die opens after the metal has hardened, the roller 52, moving up the inclined plane 53, raises the entire mechanism away from the charging chamber. While running a large aluminum die this action would prove objectionable. The lifting of a large die and the mechanism required to mount and operate it require a vast amount of power.

Fig. 6 shows another air machine of this type. Here the metal enters the die through one of its halves, very similar to the method employed in sand molds. In this respect the machine resembles the plunger type shown in Fig. 3.

This construction makes it necessary for the metal, on reaching the parting line between the die halves, to flow around an angle of 90 deg. before entering the casting cavity. This bend naturally absorbs some of the beneficial force of the flow. In this respect the air machine of Fig. 5 is better.

Due to the dies' being located beside rather than over the furnace, it does not heat so rapidly during the initial firing. The heat taken out of the first few castings remedies this shortcoming.

The operators work beside the furnace rather than over it. Consequently this machine holds more attraction for the operator than the previous one.

As the illustration shows, cores may pull in all six directions, which some castings require. If only five sides are available for cores, as is the case in Fig. 5, the holes or cavities on one side of the object cannot be cast. The subsequent machining of these parts increases the cost of the product.

Unlike the previous machine, the die while opening does not move diagonally upward, but moves back in a horizontal line. The weight of the die remains at one level, so that it requires no power to lift it.

The heavier and more complicated half of the die is the one that moves back and forth to open and close it. From a mechanical standpoint it would be more efficient to have the lighter portion move rather than the heavier, and in some instances there is a great difference in weight. If it were possible to make this change and introduce the metal as effectively as at present, the resulting combination would undoubtedly prove more satisfactory. Moving parts of reduced weight would require less power for their operation. In later machines an air cylinder has replaced the rack and pinion for opening and closing the die.

Fig. 7 is a more detailed view of the gooseneck or charging chamber of the machine shown in Fig. 6. The cross-sectional view shows it in the casting position, while the dotted lines represent the recharging position.

## DIE-CASTING ALLOYS

Back in the days when type casting monopolized the entire die-casting field, the alloy best adapted to this use was one composed largely of lead. The same holds true today.

When the industry broadened to include general castings these same lead alloys were used in the first products. But with a very

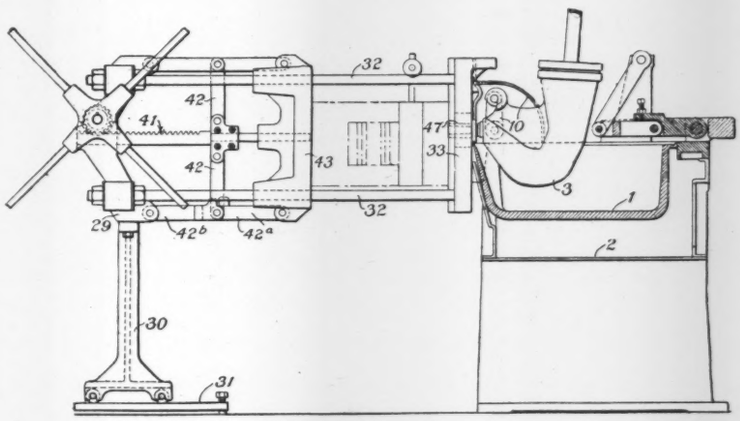



FIG. 6 ANOTHER MODERN AIR MACHINE

soft composition and a very low tensile strength, castings of these alloys could supply but a very limited field.

The addition of tin slightly improved the qualities of the castings, but its increased strength obtained was not sufficient to permit the development of die casting to any great extent. Zinc alloys offered the first opportunity to obtain castings of fair strength, which could compete with cast iron.

Since then aluminum with its strength and lightness has met with great popularity. The demands of the automotive industry have stimulated the development of alloys of this metal, and today castings from them compose the major output of the die-casting industry. Because of the beneficial effect of the rapid chilling of an aluminum alloy in a die, these castings excel those made in sand molds.

In the beginning of the industry the men that associated themselves with it knew very little about the principles of metallurgy. While they were confining their whole attention to lead and tin their ignorance was no serious drawback, but when they started to employ zinc to obtain its additional strength they encountered some serious difficulties. Their hit-or-miss methods in the alloying of metals proved very disastrous in some cases. The castings produced were satisfactory in appearance and equally so in regard to strength. However, after a period of service their bright metallic luster turned to a dull gray hue. The castings increased slightly in size, warped, and in some cases developed large cracks and even fell apart. General atmospheric conditions controlled this deterioration to a marked degree. Heat and moisture, such as from steam, greatly stimulated this action. These results gave die castings a very bad reputation. Engineers and the buying public associated this deterioration with all die castings.

A careful study of conditions has thrown a great deal of light upon these reactions. Even today the exposure to moisture of the zinc alloys in general use will quickly cause corrosion. Consequently, under such conditions some non-corrosive substance should protect them.

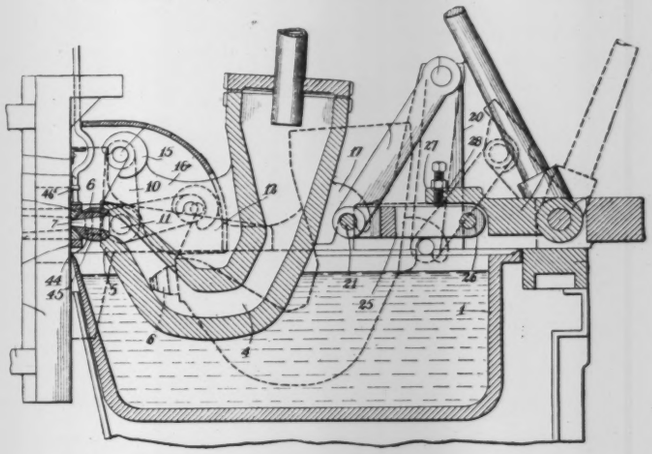



FIG. 7 ENLARGED VIEW OF GOOSENECK OR CHARGING CHAMBER OF MACHINE SHOWN IN FIG. 6

Below are given the compositions, properties, casting limitations, and applications of four alloys—one from each of the basic groups, lead, tin, zinc, and aluminum. The metal content can vary to alter the properties, but the alloys given are representative of the various groups.

### LEAD-BASE ALLOY

*Composition* (Approximate)

| | |
|---|---|
| Lead | 83 per cent |
| Antimony | 17 per cent |

*Casting Limitations*

| | |
|---|---|
| Maximum weight of casting | 25 lb. |
| Minimum wall thickness | 1/32 in. |
| Variation in size of castings per linear inch | ±0.001 in. |

*Threads*

| | |
|---|---|
| External (maximum number per inch) | 34 |
| Internal | Sometimes cast |

*Minimum Size of Holes* ..... 1/32 in.
(Depends on length of hole and size of casting)

*Draft*

| | |
|---|---|
| Cores (per inch of length and diameter) | 0.0005 in. |
| Side walls (per inch) | 0.001 in. |

*Physical Properties*

| | |
|---|---|
| Color | Gray |
| Specific gravity | 10.03 |
| Weight per cubic inch | 0.382 lb. |
| Melting point | 600 deg. fahr. |

*Chemical Properties*

Reactions:

| | |
|---|---|
| Atmosphere | Slight oxidation |
| Hydrochloric acid | Slowly soluble |
| Sulphuric acid | Not soluble |
| Nitric acid | Readily soluble |
| Organic acids | Form poisonous lead salts |

As this alloy reacts but slowly with the atmosphere it can be polished. It nickel-plates to good advantage.

*Application*

This alloy is a typical type metal. It is also one of the finest bearing metals known, where the speed and pressure are moderate. It may be used in parts where non-corrosion is a necessity. Due to its reaction with organic acids it is unsuited for food containers.

TIN-BASE ALLOY

*Composition* (Approximate)

| | |
|---|---|
| Tin | 86 per cent |
| Copper | 6 per cent |
| Antimony | 8 per cent |

*Casting Limitations*

| | |
|---|---|
| Maximum weight of casting | 20 lb. |
| Minimum wall thickness | 1/32 in. |
| Variation in size of castings per linear inch | ±0.0005 in. |

*Threads*

| | |
|---|---|
| External (maximum number per inch) | 34 |
| Internal | Sometimes cast |

*Minimum Size of Holes* 1/32 in.
(Depends on length of hole and size of casting)

*Draft*

| | |
|---|---|
| Cores (per inch of length and diameter) | 0.0005 in. |
| Side walls (per inch) | 0.001 in. |

*Physical Properties*

| | |
|---|---|
| Color | Silver white |
| Specific gravity | 7.4 |
| Weight per cubic inch | 0.266 lb. |
| Melting point | 450 deg. fahr. |
| Initial fusing point | 425 deg. fahr. |

*Chemical Properties*

Reactions:

| | |
|---|---|
| Atmosphere (under normal conditions) | No reaction |
| Mineral acids | Slow reaction |
| Alkalis | Slow reaction |
| Organic acids | No reaction |

Due to its lack of action with the elements this alloy will maintain a polish. It plates readily by the usual commercial processes.

*Application*

The chief uses for this alloy are in high-class bearings for engines and machine parts requiring a low tensile strength but high resistance to corrosion.

ZINC-BASE ALLOY

*Composition* (Approximate)

| | |
|---|---|
| Zinc | 88 per cent |
| Tin | .8 per cent |
| Copper | 4 per cent |

*Casting Limitations*

| | |
|---|---|
| Maximum weight of casting | 20 lb. |
| Minimum wall thickness: | |
| Small castings | 1/16 in. |
| Large castings | 1/8 in. |
| Variation in size of castings per linear inch | ±0.001 in. |

*Threads*

| | |
|---|---|
| External (maximum number per inch) | 30 |
| Internal | Sometimes cast |

*Minimum Size of Holes* 1/32 in.
(Depends on length of hole and size of casting)

*Draft*

| | |
|---|---|
| Cores (per inch of length and diameter) | 0.001 in. |
| Side walls (per inch) | 0.001 in. |

*Physical Properties*

| | |
|---|---|
| Color | Silver white |
| Specific gravity | 7.20 |
| Weight per cubic inch | 0.253 lb. |
| Melting point | 780 deg. fahr. |
| Initial fusing point | 275 deg. fahr. |
| Tensile strength | 16,000 lb. per sq. in. |
| Elongation | 2 per cent |
| Compressive strength | 27,500 lb. per sq. in. |
| Hardness (Brinell) | 64.6 |

*Chemical Properties*

Reactions:

| | |
|---|---|
| Moist atmospheric conditions cause | Corrosion |
| Mineral acids | Soluble |
| Alkalis | Soluble |

As this alloy may be readily plated with the common plating materials, such as nickel, copper, brass, silver, or gold, these substances may be used to protect it from corrosion. It may also be soldered readily.

*Application*

This alloy deteriorates due to a slight crystalline growth. It will not stand sudden shocks or severe strains. A few of its uses are in castings of phonograph parts, magneto frames, calculating machines, automobile accessories, etc.

ALUMINUM-BASE ALLOY

*Composition* (Approximate)

| | |
|---|---|
| Aluminum | 92 per cent |
| Copper | 8 per cent |

*Casting Limitations*

| | |
|---|---|
| Maximum weight of casting | 10 lb. |
| Minimum wall thickness: | |
| Small castings | 1/16 in. |
| Large castings | 1/8 in. |
| Variation in size of castings per linear inch | ±0.0025 in. |

*Threads*

| | |
|---|---|
| External (maximum number per inch) | 20 |
| Internal | Very rarely cast |

*Minimum Size of Holes* 3/32 in.
(Depends on length of hole and size of casting)

*Draft*

| | |
|---|---|
| Cores (per inch of length and diameter) | 0.015 in. |
| Side walls (per inch) | 0.005 in. |

*Physical Properties*

| | |
|---|---|
| Color | Silver white |
| Specific gravity | 3.003 |
| Weight per cubic inch | 0.115 lb. |
| Melting point | 1150 deg. fahr. |
| Initial fusing point | 1025 deg. fahr. |
| Tensile strength | 21,000 lb. per sq. in. |
| Elongation | 1.5 per cent |
| Compressive strength | 38,000 lb. per sq. in. |
| Hardness (Brinell) | 60.5 |

*Chemical Properties*

Reactions:

| | |
|---|---|
| Atmosphere | No reaction |
| Alkalis | Soluble |
| Nitric acid | Not soluble |
| Hydrochloric acid (dilute) | Soluble |
| Hydrochloric acid (concentrated) | Slightly soluble |
| Sulphuric acid (dilute) | Soluble |
| Sulphuric acid (concentrated) | Slightly soluble |
| Organic acids | Little reaction |

This alloy will retain a luster after polishing for a long period.

*Application*

Aluminum alloys are used most widely, chiefly on vacuum cleaners, phonographs, cameras, and automobiles. As the products formed with organic acids are harmless, this alloy may be used in food containers.

In alloying or melting these metals they react with a certain amount of the cast iron of the containers. In the case of a copper-aluminum alloy similar to the one described above, the iron combines with the aluminum, forming $FeAl_3$. Certain non-metallic compounds also form, and small pieces of scale from the containers are washed into the alloy. Small particles of these compounds studded through the casting form what are known as "hard spots," because they are harder than the alloy itself. Ordinarily these compounds are not objectionable. In some cases, however, it is difficult to machine castings, as the hard spots dull the tools so quickly. At times the entire spot will tear out, leaving a small jagged hole in the casting.

The greater the amount of iron in the metal, the higher will be its melting point. Greater sluggishness accompanies these increases, so that the mold fills with more difficulty. The usefulness of the alloy becomes questionable when its iron content is in excess of 3 per cent.

In cases where the volume of the casting greatly exceeds that of the gate, there is very little trouble due to the presence of iron. However, in the case of small castings that require a long gate of good size, so that the volume of the gate greatly exceeds that of the casting, the increased presence of iron aggravates this condition. In this case a large percentage of the metal cast (the gate) returns to the pot for remelting. In this way the metal dissolves more cast iron on each return to the melting pot.

## DIES

Every new form of casting means a new die, and every die has its problems. Consequently the design and construction of dies plays an important part in the industry.

The sections of Fig. 8 show a three-impression die for a gearwheel. These views show the general method of constructing and of operating the die. Fig. 12 (*B*) pictures this casting.

In Fig. 8*a* the full-line and broken-line portions represent the ejector die; the dash-and-dot line, the die base; the dash-and-double-dot line the cores and core mechanism; and the double-dash-and-dot line the ejector pins and ejecting mechanism. Referring to Fig. 8*c*, a turn of the core pinion removes the cores, while a turn of the ejector pinion ejects the casting. When the die closes after the ejection of the casting the pressure of the cover die upon the four surface pins pushes the ejector plates, and consequently the ejector pins, down into position.

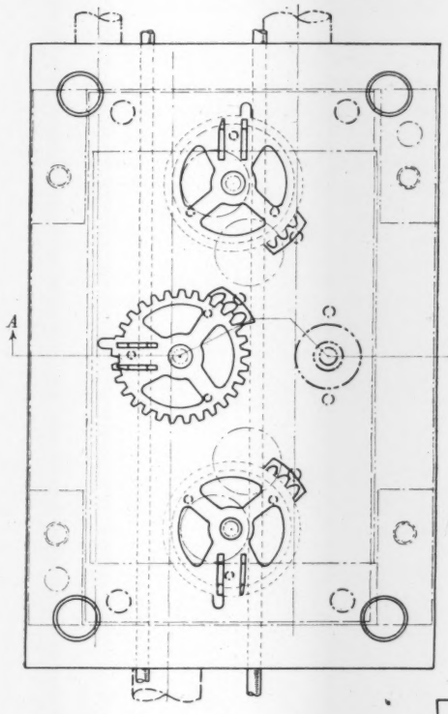



FIG. 8*a* EJECTOR DIE FOR GEARWHEEL

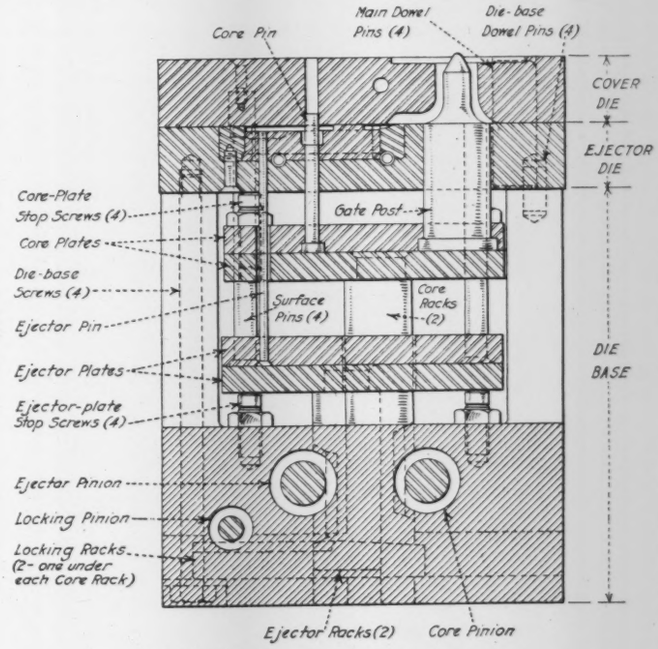



FIG. 8*c* SECTION *A-A* OF FIGS. 8*a* AND 8*b*

In this die the casting is entirely within the ejector half. Due to the shrinkage of the casting about the three pads that form the openings in the gears, about the center core, and about the gate post, the casting will cling to this half of the die each time the die opens. Consequently but one ejecting mechanism is needed, which is in this half. For this reason this die is called the "ejector die." The other half of the die merely covers it, completing the impression, so that it is called the "cover die." These halves of the die may be likened to the cope and drag of a sand mold.

The parting line between the halves of the die must contain the gate and the vents. The location of the gate should afford the metal easy access to all portions of the impression. Its position should be such that any irregularities caused by its removal from the casting will not jeopardize the appearance or usefulness of the part. In this case the metal enters through the ends of the gear teeth.

Several methods are used to vent the castings. It is not only necessary for the air in the impression to escape but also for that in the gate and the casting chamber of the machine to pass through the vents. In some cases shims about five thousandths of an inch in thickness are placed about the four dowel pins to form the vent. These shims hold the two dies apart, thus enabling the air to escape over the entire die surface. In other cases grooves ground in the surface of the die and extending from the impression to the edge of the die, perform the same function. These grooves are from $^1/_2$ in. to 2 in. in width, depending on the size of the casting needing venting. On white-metal dies they are about five thousandths of an inch deep and on aluminum dies about seven or eight thousandths.

Sometimes in a deep cavity of the die a square ejector pin placed in a round hole will suffice to vent this portion of the impression. The space around the ejector pin forms the vent. The motion of the pin while ejecting the casting is sufficient to keep the hole free from fins.

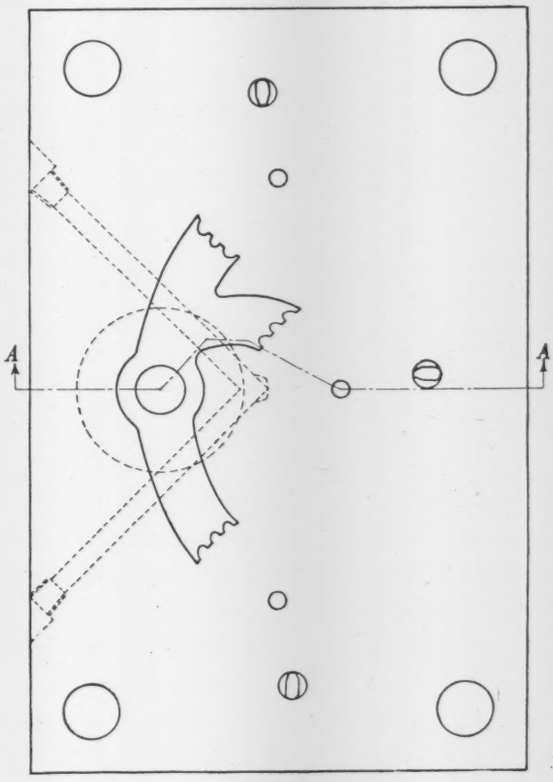



FIG. 8*b* COVER DIE FOR GEARWHEEL

Figs. 9 and 10 illustrate two additional methods of gating and venting dies, both of which are extremely popular. In Fig. 9 the metal flows through the gate into one end of the impression while the air passes out the other end. This makes an ideal condition in that there is one continuous, unobstructed flow of metal. A simple turning operation will remove the gate economically.

Fig. 10 shows another casting with the metal entering from the under side. In this case the metal shoots upward over the core. The location of the vents at the extreme right of the impression affords a direct flow of the metal from one side of the casting to the other. In this case a milling operation removes the gate.

When locating the gate, the vents, and even the parting line of the die, we must consider the removal of the fins formed by these various members. These operations should naturally be as simple as possible. In a great many instances the gate may be broken from the casting. Afterward a file removes the remaining roughness and the venting fins. In some cases a simple machine operation, as noted above, may facilitate this cleaning. In the case of the die in Fig. 8, by pressing the gear through a shaving die that matches the periphery of the casting, all of the fins are removed in one quick and simple operation. This method is very popular where the number of castings on order warrants the construction of such a die.

In some cases it is necessary to remove the imprint of the ejector pin from the surface of the casting. It is impossible to keep the surface of an ejector pin flush with the surface of the impression at all times. The unequal expansion of various parts of the die—ejector pins, die blocks, etc.—due to the heat of the casting metal, makes these surfaces vary. These marks will always be visible on the rough casting, and in most instances they are not objectionable. However, if the pins are on a surface which will receive a high polish,

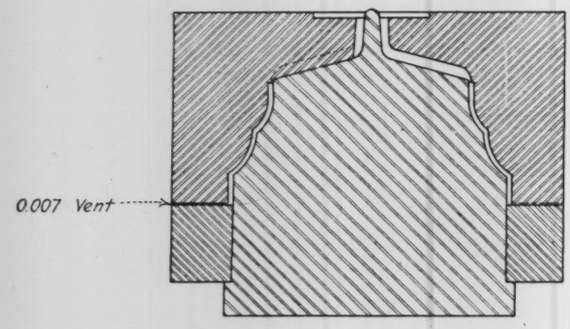



Fig. 9

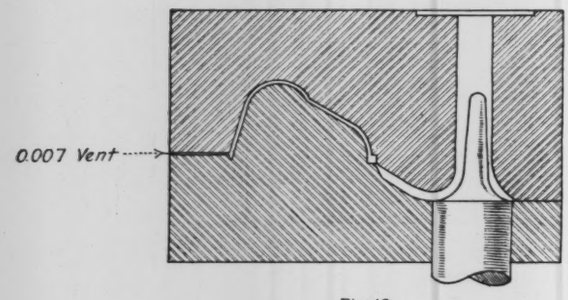



Fig. 10

FIGS. 9 AND 10 METHODS OF VENTING AND GATING DIES

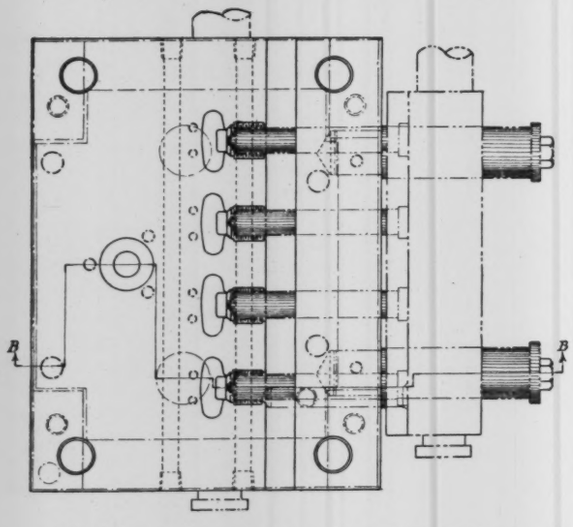



FIG. 11a EJECTOR DIE FOR A SCREW FAUCET PLUG

by placing them five or ten thousandths of an inch below the surface of the die we may grind off the bosses thus formed on the casting before the polishing operation. If the pins are on a surface that will fit against some other part when assembled, they may extend five or ten thousandths of an inch above the die. The resulting depression in the castings will not affect the fit between the two parts in any way.

The constant operation of the die creates a great deal of heat within the die steel. All the excess heat of each casting passes into the die itself. This heat includes the latent heat of solidification plus the heat it is necessary for the casting to lose in order to reach a stable temperature. Water passing through holes drilled in the die block removes this heat. The location of these water channels at strategic points of the die keeps all parts of the steel at as nearly the same temperature as possible.

If this cooling functions properly, the rate of production depends very largely on the mechanical limitations of each particular die. The large, complicated ones, with a great many cores to operate, are of necessity slow. The small, light ones are correspondingly fast.

Figs. 11*a*-11*c* show a four-impression die for a faucet screw plug. These sketches employ the same line scheme as those of Fig. 8. In this die the core mechanism operates in a horizontal plane. Only the ejecting mechanism functions from the die base. In dies of this type the ejector safety, as shown, adds materially to the dependability of the die. Any attempt to eject before pulling the side cores places all of the premature effort on the safety rather than on the more fragile ejector pins and castings. This safeguard elimi-

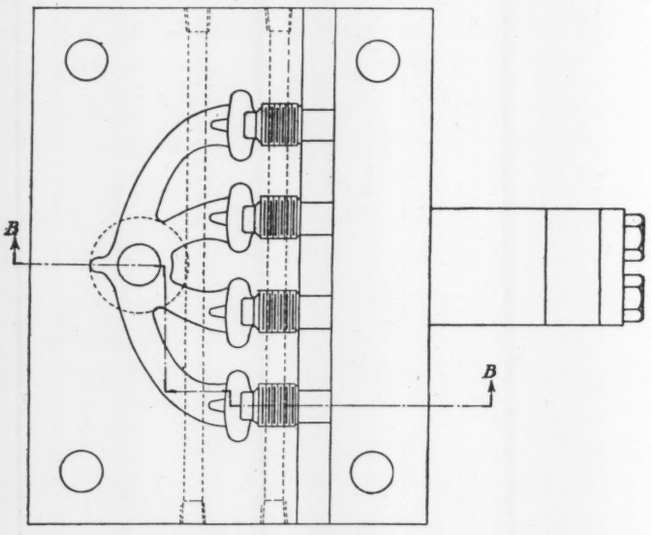



FIG. 11b COVER DIE FOR A SCREW FAUCET PLUG

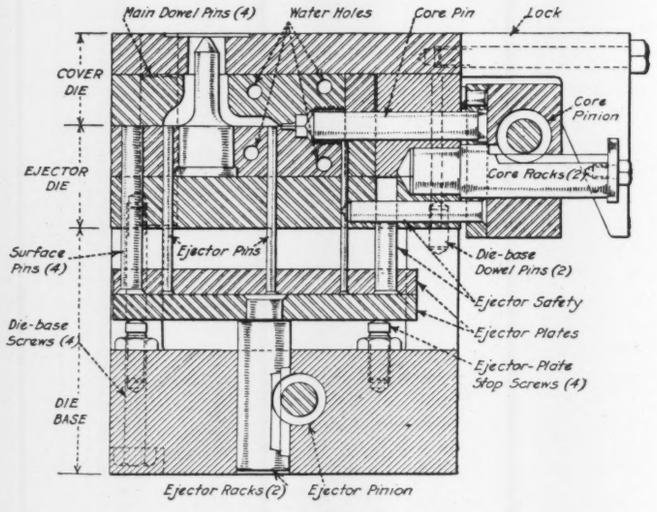



FIG. 11c SECTION *B-B* OF FIGS. 11*a* AND 11*b*

nates a great many breakdowns. Fig. 12(*A*) shows the finished casting.

In the case of threads, the logical way to clean them is by means of a stamping die. In order to remove the castings from the die, it is necessary to part the latter along the center line of the thread diameter, having half the thread in each half of the die. In this way there will be a portion of the thread diameter across the parting line of the die. As small portions of the venting fins and other dirt very often stick to the surfaces of the dies, and as there is a certain amount of play in the operating linkage of the casting machine, the dies very often stay open a few thousandths of an inch. This amount varies from about five thousandths of an inch in small, simple dies to

about ten thousandths in larger and more complicated ones. Such opening of the dies causes the dimension across it to be a corresponding amount larger than a completely closed die actually measures. In most cases this tolerance is entirely satisfactory. However, if the part demands a closer tolerance, the running of a threading die over the threads will bring them to the desired size. As these threading dies take but a slight shaving cut in this case, they operate at a high speed on an automatic threading machine.

The two dies illustrated in Figs. 8 and 11 are comparatively simple. However, the larger and more complicated ones employ the same principles of construction and operation.

In most cases, when designing a part to be die cast, it is desirable to eliminate all undercuts. By this is meant having each core removed from the casting and also the casting removed from the die, intact, and in one straight line. This is the simplest manner in which to produce die castings.

In some cases where undercuts are essential, a large number of castings will justify the additional expense of collapsible cores and other complicated die constructions. Figs. 13, 14, and 15 illustrate three castings in this category.

Fig. 13 shows a phonograph tone arm containing a U-shaped hole or sound passage. In the first stage the casting is complete with the exception of the upper portion. Here a hole through the bend in the tube permits the exit of the core that forms the cross-member of the U-shaped passage. This casting, used as an insert in a second die, together with the metal added at the bend, which closes the temporary hole, forms the completed casting.

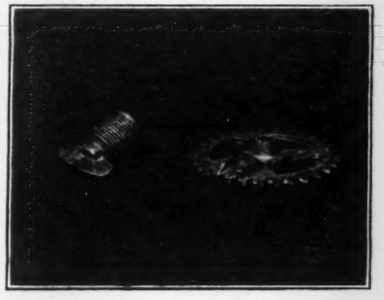


FIG. 12 SCREW PLUG AND GEAR

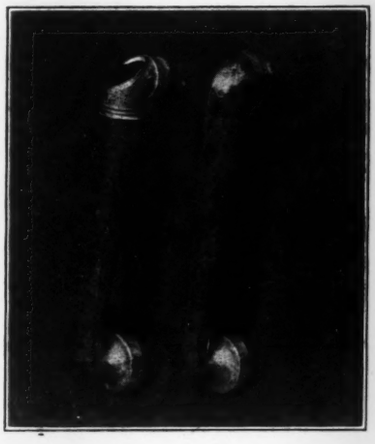


FIG. 13 TWO STAGES IN THE CONSTRUCTION OF A PHONOGRAPH TONE ARM

Fig. 14 shows a syrup-pump body which requires four stages for its production. The first stage, as indicated at $A$, consists of a section of tin-plated copper tubing. This tubing used as an insert in a die, forms the nozzle of the pump. This casting, bent as at $B$, forms an insert for a second die which produces the completed body as at $C$. $D$ shows the body cut in two.

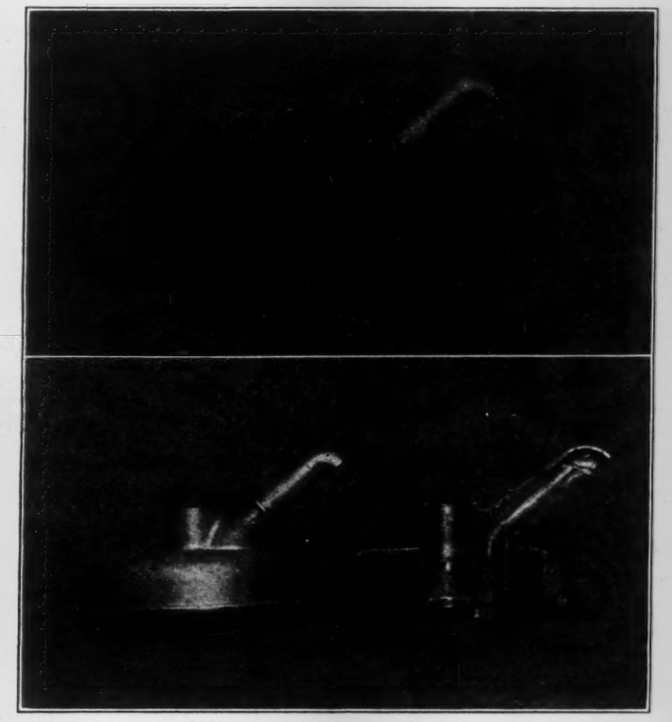


FIG. 14 SYRUP-PUMP BODY

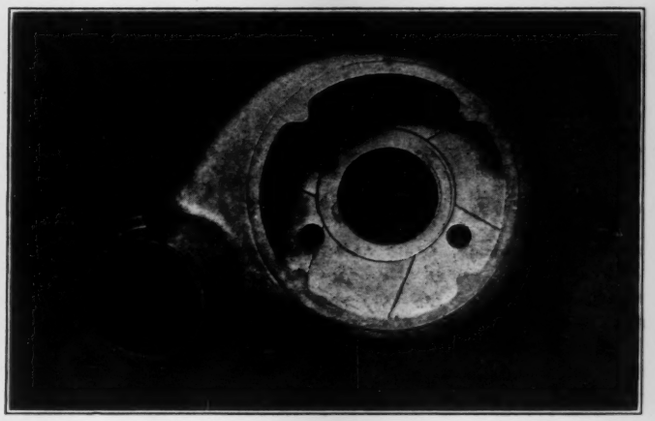


FIG. 15 VACUUM-CLEANER-FAN CASE

The casting shown in Fig. 15 is a vacuum-cleaner-fan case. The lines shown on the inside of this part are the result of the fins formed by the split lines in the core. They indicate its construction which after the removal of the round center section, permits each individual piece to be pulled toward the center of the casting for extraction.

What kind of material will make the best dies? How long will these dies last? These are two questions which metallurgists are endeavoring to answer.

In the case of a die for lead or tin, a good grade of carbon machine steel will last indefinitely. The heat that passes from the casting into the die is not great enough to affect the steel.

In the case of zinc castings, which possess more heat for absorption, the additional heat strains the steel more severely. After about twenty-five thousand castings a carbon-steel die develops small surface cracks which are the result of the thermal fatigue of the steel.

Aluminum with its higher melting point causes a more drastic action. Using a special alloy steel, properly heat treated, these dies develop heat cracks after from 5000 to 15,000 castings, depending on the size and shape of the casting. However, the useful life of these dies varies from 25,000 to 200,000 casting operations.

Fig. 16 illustrates what takes place to cause these cracks. The heat from the molten metal when injected into the die raises the entire impression surface $A$ to within a few degrees of its own temperature. This increase in temperature tends to make the die expand in all three dimensions. The tendency for a small portion of the steel block would be to expand to a shape similar to the dotted lines at $x$ in the figure. The cooling effect of the water lines at $B$ keeps the under portion of the die within a few degrees of the temperature of the water. These portions do not expand. As the small element of steel shown at $x$ cannot expand horizontally because it is in the center of a steel block, it expands to a shape similar to the dotted lines at $y$. The upper portion of the element at $C$ will be under a compression strain. After the casting leaves the die and before the next charge of metal enters, the impression surface of the die cools somewhat, relieving this compression strain.

## CAUSES OF HEAT CRACKS IN DIES

These periodic strains occur many times an hour. Eventually the impression surface becomes permanently deformed and the elements of steel, during the cooling period of the cycle, tend to

assume a shape as illustrated at $z$ in Fig. 16. For a while the tensile strength will hold the element to the shape at $y$. Finally the surface gives way, the elements assuming a shape similar to that at $z$. This leaves a crack between this element and the adjacent one. These fine hairline cracks spread over the surface, forming a network of these crevices. The succeeding shots of molten metal wedge their way into these cracks, aggravating greatly the thermal action.

The presence of these cracks in the die causes corresponding ridges in the casting. Such dies may continue in use until the cracks are so large and numerous that it is not economical to remove the resulting fins from the castings. Upon the size and bulk of the casting depend the rapidity with which the die will crack. The size to which these cracks may safely grow depends upon the duties which the castings will perform.

Fig. 17 shows the top of a ventilating lid for the cowl of an automobile. This casting is the sixteen-thousandth produced from its die. A close examination will show slight ridges, which are due to the fatigue cracks in the die. The upper part of Fig. 18 shows the die block that produced this casting, and the lower part a portion of the same die block enlarged five times. This view clearly shows the fatigue cracks. The white lines are aluminum that has forced its way into the cracks. This thermal-fatigue failure will continue until the block becomes useless.

The problem of eliminating this fatigue failure seems to require a die material whose elastic limit is high enough to prevent the temperature changes from stressing the material sufficiently to cause permanent deformation. Such conditions may be had by raising the elastic limit, decreasing the coefficient of expansion, or by a combination of both.

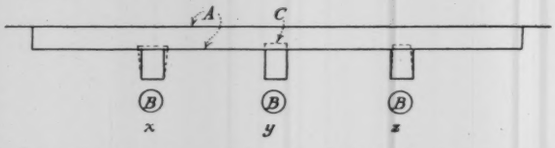



FIG. 16 DIAGRAM ILLUSTRATING CAUSES OF HEAT CRACKS IN DIES

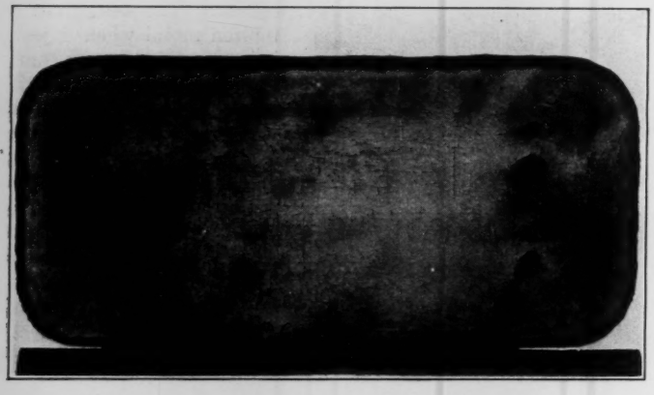


FIG. 17 AUTOMOBILE VENTILATOR

If these changes could be carried far enough, brass castings could be economically die cast. The heat action of the brass is so drastic on the present die steels that their lives are very short. The castings are successful, but the cost of the die prohibits the production of so few castings. If a substance could be found that would produce an indefinite number of aluminum castings without the annoyance of thermal fatigue, the same substance would commercially produce brass castings. At present no such substance is known and the prospect of finding one in the near future seems very remote.

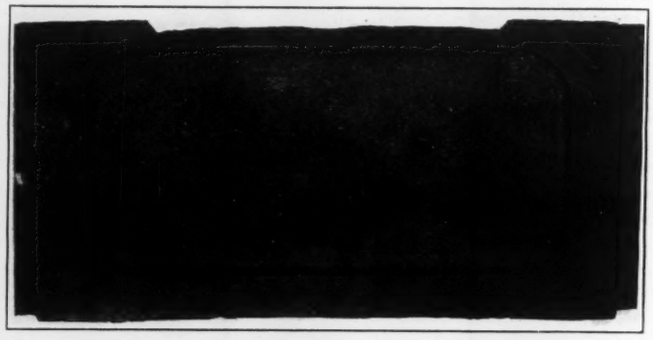
## DEFECTIVE CASTINGS

Blowholes (gas bubbles or pockets in the casting) are a very common fault. These imperfections reduce the strength of the part and, if they occur on the surface, mar its appearance. The causes of these defects may be classified as dross or slag, porous metal, improper venting of the die, improper gating of the die, and "piping."

A certain amount of slag finds its way into the castings. However, failures due to its presence are so few that they may be disregarded.

Metals in the molten state absorb a certain amount of gases from the air. These gases pass into solution in the metal. When the metal solidifies, the gases, as they are not soluble in the solid material, go out of solution, forming microscopic pores. Due to their small size they are not objectionable. Naturally, however, in order that the material may be as solid as practicable, it is desirable to have as few of these pores as possible.

Experiments show that the hotter the metal while melting the longer it remains molten, or the hotter it is during the casting operation the more porous will be the solidified material. It is therefore desirable to melt the alloys at as low a temperature as



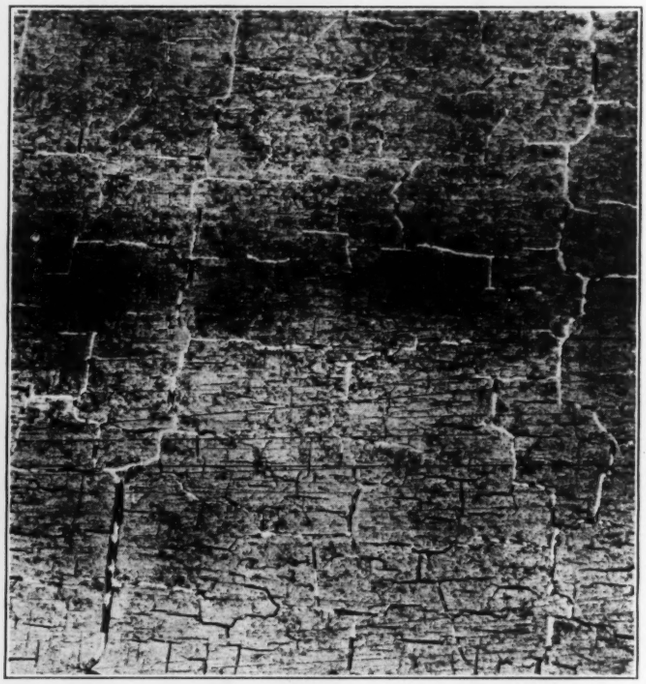


FIG. 18 ABOVE: DIE BLOCK FOR MAKING AUTOMOBILE VENTILATOR SHOWN IN FIG. 17. BELOW: PORTION OF DIE BLOCK ENLARGED

possible, keeping them melted as short a time as possible. Lastly, it is desirable to cast the material at the lowest temperature at which it will properly flow into the mold. By these precautions we decrease the porosity to some extent and consequently increase the strength.

If the die impression has not a proper vent, part of the air in it becomes entrapped in the casting. In some cases the location of the gate causes the metal to flow through a thin section of the impression into a larger portion of the part. The metal in the thin portion of the casting is very apt to solidify before the larger portion fills. This leaves the large section of the part incompletely filled. In the section dealing with dies several cases of proper gating have been illustrated.

"Piping," or the formation of shrinkage voids, as in other methods of casting, usually starts in the gate and tends to extend to the casting. If pressure is maintained in the casting chamber so that new metal can be forced into the die as that which is already in it shrinks, the chances of "piping" within the castings, will be reduced to a minimum.

Study and experiments conducted with that purpose in view will eliminate the objectionable features of these defects from practically any casting.

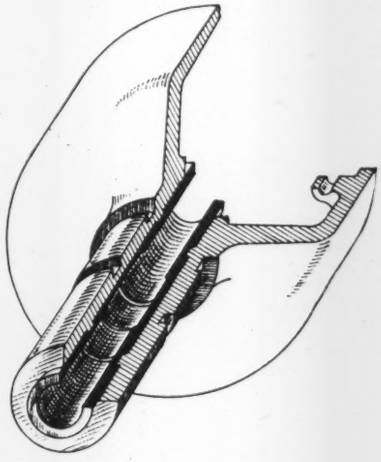


FIG. 19 BASE CASTING FOR AN IGNITION UNIT

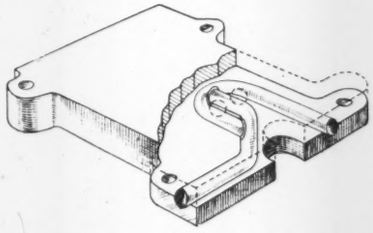


FIG. 20 VALVE BLOCK FOR A VACUUM WINDSHIELD CLEANER

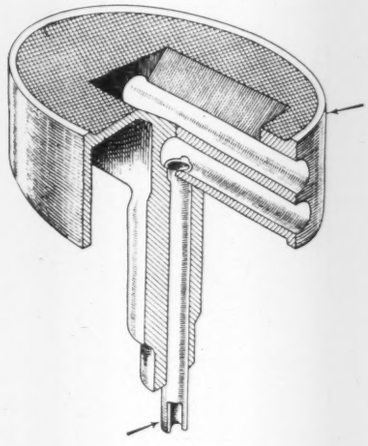


FIG. 21 AUTOMOBILE DASH GAGE

## INSERTS

Certain portions of many castings are required to have specific properties other than those of the die-casting materials. A material possessing the desired qualities, if placed in a die, will form an integral part of a casting after the molten metal solidifies. In this way portions of the casting may be provided with local properties that the die-casting alloys cannot furnish.

In Fig. 19 two bronze inserts, placed end to end, form a bearing through the center of the casting. These inserts furnish a far better bearing surface than would be the case if the aluminum were used throughout.

Figs. 20, 21, and 22 illustrate the use of tubing. In Figs. 20 and 22 the tubing forms a passage too small and crooked to be cored with a steel core. The end of the tube projecting from the lower portion of the casting in Fig. 21 brazes or solders more readily than the alloy composing the remaining part of the gage.

The soft-iron pole pieces, shown at *A* in Fig. 23 provide certain magnetic properties that the casting itself does not possess. The inserts at *B* of this same casting, after being drilled and tapped, furnish a more durable thread than would the zinc-base alloy of the casting.

Through the use of inserts similar to the ones illustrated the usefulness of the industry far exceeds the possibilities of the products made from the unaided casting materials.

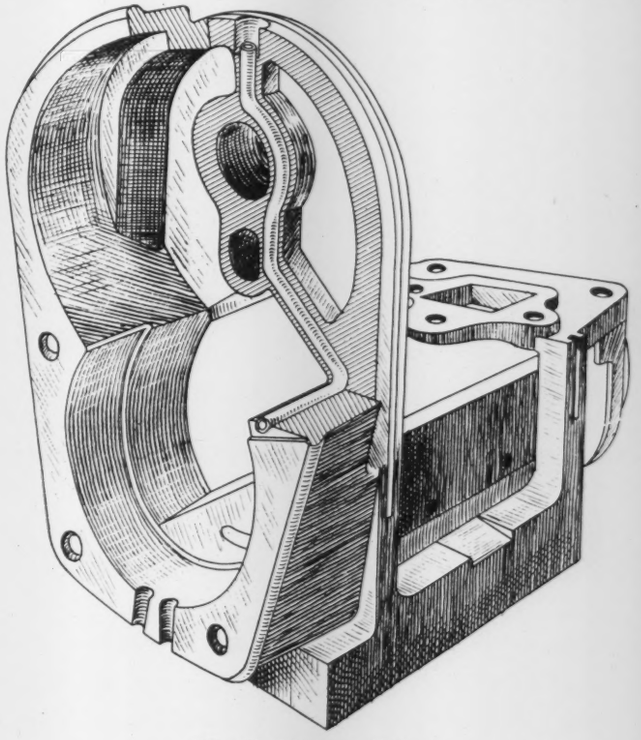


FIG. 22 MAGNETO FRAME

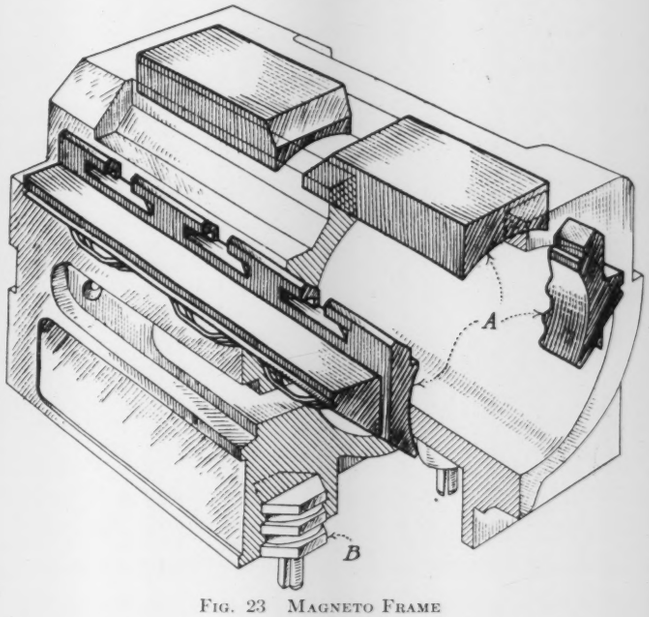



FIG. 23 MAGNETO FRAME

## CONCLUSION

The construction of the die places a high initial cost upon the product. If only a few castings are required, the cost of the die will be prohibitive. However, if a large number of castings is desired, the cost per part dwindles to a very low figure. Starting with this fundamental principle, die casting is most advantageous where many castings are required.

The process is most economical where its close casting tolerances eliminate machine operations that would be necessary on castings from other processes. Thus the money saved on a large order more than compensates for the cost of the die.

This process cannot compete with sand castings that require little or no machining. In a similar way it cannot compete commercially with simple sheet-metal stampings and simple screw-machine parts. The rapidity with which automatic and semi-automatic machines turn out these two classes of products reduces their cost to a minimum. In general, the initial cost of the die for die casting the same parts eliminates this method for those types of work.

The products of die casting are constantly becoming larger and more complicated. At the present time the die casting of automobile body frames is contemplated. In the near future the industry will probably enlarge to include such castings as crankcases. Such parts will require the construction of new and larger casting machines.

Today the die castings of a few years ago seem very simple. No doubt many of the castings of the present will assume a similar aspect to the engineers of a few years hence. Die casting is a young, growing industry whose possibilities can only now be surmised.

# The Rotating Disk

By E. L. THEARLE,[1] ITHACA, N. Y.

*In the classical theory of the eccentric rotating disk as presented by Foppl, the center of gravity of the eccentric disk is assumed to lie always in the plane of the deflected shaft, and angular acceleration is neglected entirely. The solution based on this theory leads to a critical speed, at which the shaft deflection becomes infinite. Since a mechanism such as a De Laval turbine would become inoperative under such conditions, the idea of "violent vibration" at the critical speed is introduced. The author has attempted to develop a more rational theory by writing down the equations of motion of the particles of the disk in more general form and introducing the idea of possible angular acceleration. The maximum shaft deflection at the critical speed is found to have a finite value.*

IN THE DESIGN of small, high-speed, single-stage steam turbines of the De Laval type, and perhaps other high-speed machinery, the whirling-disk problem becomes of considerable importance. This problem deals with the behavior, at high rotative speeds, of a disk or load mounted on a comparatively slender and flexible shaft which is supported at its ends in bearings. The most common solution of this problem is that due to Föppl, which is outlined below.

### Classical Theory of the Eccentric Rotating Disk

Referring to Fig. 1, the disk $D$ is shown mounted on the shaft $AB$ which is supported in bearings at its ends $A$ and $B$. In the manufacture and mounting of such a disk on a shaft it is impossible to place the center of gravity of the disk exactly on the axis of the

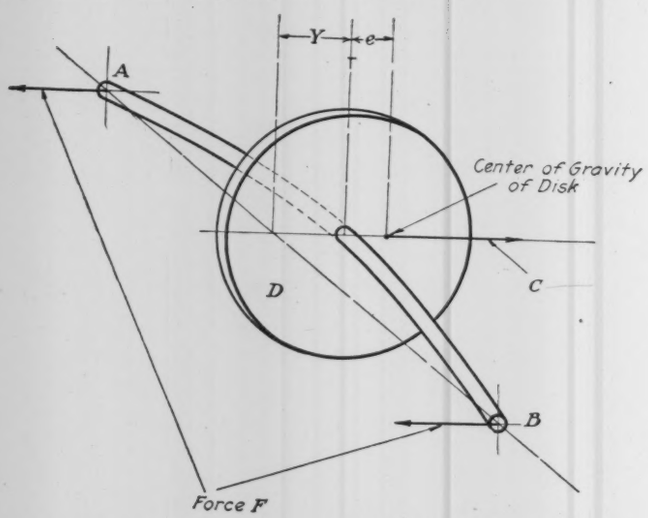



Fig. 1 Diagram Illustrating the Deflection of an Eccentric Rotating Disk

shaft or the axis of rotation. The center of gravity of the disk will always be off center by an amount $e$, where $e$ represents the distance between the center of gravity of the disk and the center line of the bearings when the disk is at rest, or the original eccentricity of the disk, assuming the shaft to be vertical. Thus as the shaft and disk are rotated the center of gravity of the disk moves in a circular path about the axis of rotation $AB$. In order to accelerate the disk toward its axis of rotation a force must be applied to it, acting in this direction. This force, $F$, transmitted to the disk by the shaft is induced by the deflection of the shaft and is proportional to this deflection $Y$. Or it may be stated that the deflection of the shaft, $Y$, is proportional to the force which produces it. Thus

$$Y = KF \quad [1]$$

where $K$ is a constant depending upon the proportions of the shaft. Equation [1] may be written

$$F = \frac{Y}{K} \quad [2]$$

At any rotative speed $\omega$ the center of gravity of the disk then moves in a circular path of radius $(Y + e)$ about its axis of rotation.

Most authors prefer to imagine a "centrifugal force" $C$ (Fig. 1) produced by this rotation of the center of gravity of the disk, and this centrifugal force

$$C = \frac{W}{g}(Y + e)\omega^2 \quad [3]$$

where $W$ = weight of disk
$g$ = acceleration due to gravity = 386 in. per sec. per sec.
$(Y + e)$ = radius of curvature of path
$\omega$ = angular velocity in radians per sec.

But the "centrifugal force" $C$ is that which produces the deflection of the shaft, hence

$$C = F \quad [4]$$

which, by substituting values of $C$ and $F$ from Equations [2] and [3], may be written

$$\frac{W}{g}(Y + e)\omega^2 = \frac{Y}{K} \quad [5]$$

Solving this equation for the deflection $Y$,

$$Y = e\frac{\omega^2}{\frac{g}{WK} - \omega^2} \quad [6]$$

The quantity

$$\frac{\omega^2}{\frac{g}{WK} - \omega^2} = \frac{Y}{e} \quad [7]$$

may be called the coefficient of deflection.

From Equation [6] it is seen that when

$$\omega^2 = \frac{g}{WK} \quad [8]$$

the shaft deflection $Y$ becomes infinite. The particular value of $\omega$ which establishes this condition in Equation [6] is called the "critical speed" and is represented by $\omega_c$. Thus

$$\omega_c = \sqrt{\frac{g}{WK}}$$

Equation [6] may then be written

$$Y = e\frac{\omega^2}{\omega^2_c - \omega^2} \quad [9]$$

or

$$\frac{Y}{e} = \frac{\omega^2}{\omega^2_c - \omega^2} \quad [10]$$

where $e$ is the original eccentricity of the load or center of gravity of the disk.

Equation [10] is shown graphically in Fig. 2. This equation indicates that as the speed is increased the shaft deflection increases until, at the critical speed, it becomes infinite. With further increase of speed the shaft deflection is negative and decreasing, until, as the speed becomes very large, the shaft deflection approaches a value of $-e$, or the disk approaches a condition of rotation about its center of mass.

A comparison of this solution with experience indicates that the solution is incomplete, for if the shaft deflection becomes infinite at the critical speed, it would be impossible for such a mechanism to operate at speeds above the critical. This discrepancy has been

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explained by making the statement that *at* the critical speed the disk and shaft tend to come into a state of violent "vibration," and that in bringing the disk up to a speed above the critical, the critical speed "is passed through so quickly that the disk does not have 'time' to acquire a dangerous amplitude of vibration." This latter statement involves the idea of angular acceleration, which has been neglected entirely in the above solution.

It is evident that in some way or another the center of gravity of the disk moves around the shaft, at the critical speed, until it lies inside the shaft, or between the shaft and the axis of rotation.

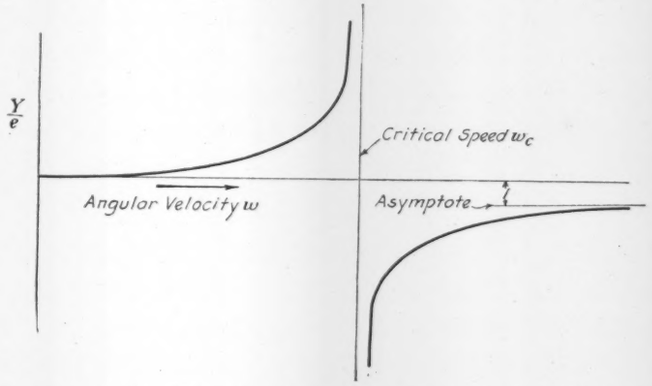


FIG. 2 DIAGRAM ILLUSTRATING RELATIONSHIP BETWEEN THE ECCENTRICITY OF A DISK, ANGULAR VELOCITY, AND SHAFT DEFLECTION

This is not indicated in the above solution because the assumption is made that the center of gravity of the disk at all times lies in the plane of the deflected shaft.

Equation [6] supplies a means of determining the critical speed, but it gives no means of evaluating the forces which must be resisted by the shaft *at* the critical speed, when these forces are a maximum.

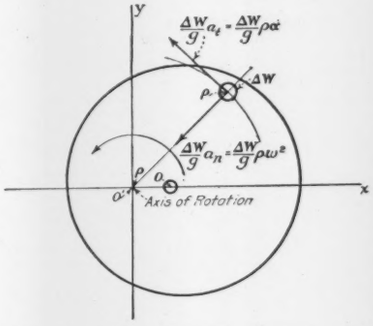


FIG. 3 DIAGRAM ILLUSTRATING THE RADIAL AND TANGENTIAL COMPONENTS OF ACCELERATION OF A PARTICLE OF THE ECCENTRIC DISK AT ANY INSTANT OF TIME

## A NEW THEORY OF THE ECCENTRIC ROTATING DISK

The following solution of the whirling-disk problem is presented with the possibility in view of its furnishing a more satisfactory means of explaining the behavior of the mechanism at its critical speed, and the more remote possibility of evaluating the forces resisted by the shaft at the critical speed. It is by no means a conclusive proof that the motion of the disk is pure rotation, since this condition must be assumed in order to obtain a solution. The results, however, may serve as a rational basis for proportioning the rotating parts.

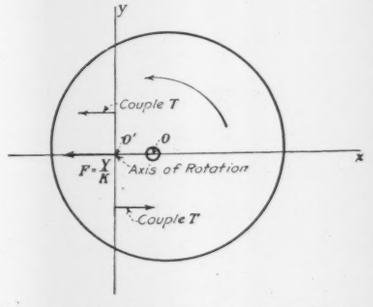


FIG. 4 FORCES AND COUPLES IN PLAY

Figs. 3, 4, and 5 represent the disk mounted on its shaft at $O$, at any instant of time, rotating about the axis $O'$ with an angular velocity $\omega$ and an angular acceleration $\alpha$. Its motion is assumed to be pure rotation about the fixed axis $O'$, which in this case is the center line of the supporting bearings. In order to simplify the following work, the shaft and axis of rotation are assumed to be vertical. The mass of the shaft is assumed to be negligible as compared to that of the disk.

An $x$-axis is arbitrarily chosen perpendicular to the axis of rotation and lying in the plane of the shaft deflection. The $x$-axis intersects the axis of rotation at $O'$ and the shaft at $O$. The $y$-axis is chosen perpendicular to the $x$-axis and to the axis of rotation, intersecting the latter at $O'$. The origin then lies on the axis of rotation, and the center of the shaft (at the disk) always lies on the $x$-axis. It is evident that these two ($x$- and $y$-) axes rotate with the disk.

The real forces acting on the disk as a whole are represented in Fig. 4. These are the forces representing the couple $T$ and the force $F$ exerted on the disk by the shaft, which are the *only* real forces exerted on the disk (as a whole) that have components in the $x$-$y$ plane. Since the force $F$ is induced by the deflection of the shaft, this force lies in the plane of the shaft deflection, perpendicular to the shaft, and therefore along the $x$-axis. This force $F$ is proportional to the shaft deflection $Y$ (Fig. 5) which induces it, hence

$$F = \text{a constant times } Y$$

or

$$F = \frac{Y}{K} \qquad [11]$$

where $K$ is a constant depending upon the proportions of the shaft.

In Fig. 3, $\Delta W$ represents the weight of a small particle of the disk. The acceleration of this particle may be represented by two components: $a_n$ normal to the path of the particle and $a_t$ tangent to its path. Also, the force which produces this acceleration may be represented by two components: $\frac{\Delta W}{g}a_n$ normal to, and $\frac{\Delta W}{g}a_t$ tangent to, the path of the particle.

Since

$$a_n = \rho\omega^2 \qquad [12]$$

and

$$a_t = \rho\alpha \qquad [13]$$

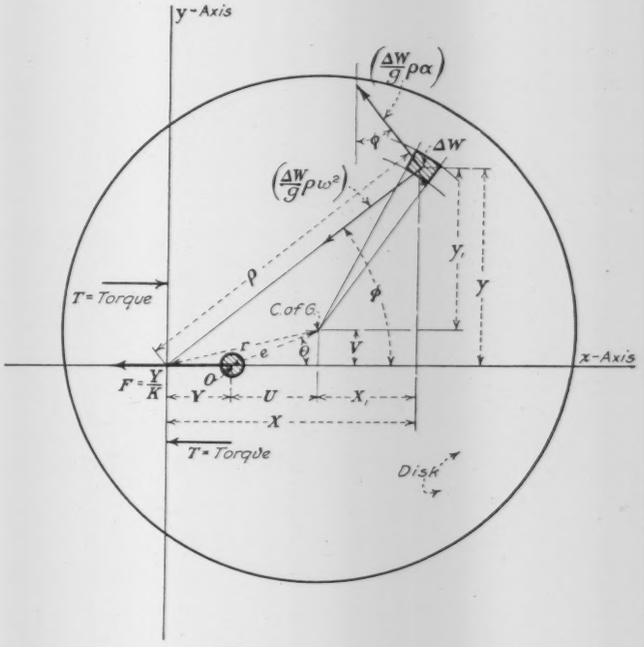


FIG. 5 FORCES ACTING ON A PARTICLE AT ANY INSTANT

(Shaft assumed to be vertical; unit of length employed in discussion = 1 in., making $g = 32.2 \times 12 = 386$ in. per sec. per sec.)

the above components of force may be written

$$\frac{\Delta W}{g}a_n = \frac{\Delta W}{g}\rho\omega^2 \qquad [14]$$

and

$$\frac{\Delta W}{g}a_t = \frac{\Delta W}{g}\rho\alpha \qquad [15]$$

If, as assumed, the motion of the disk is one of rotation about the axis $O'$, the resultant force for each and every particle of the

disk may be expressed by the components $\frac{\Delta W}{g}\rho\omega^2$ directed toward the center of rotation and $\frac{\Delta W}{g}\rho\alpha$ tangent to the path.

Moreover the vector sum or resultant of all these forces, extended to include all particles of the disk, must be equal to the resultant of the real forces applied to the disk as a whole. It follows that the $x$ and $y$ components of the system of forces represented in Fig. 4 may be equated to the $x$ and $y$ components, respectively, of the system of forces represented for one particle in Fig. 3, summed to include the entire disk. Or, from Fig. 5, summing components of force parallel to the $x$-axis,

$$F = \frac{Y}{K} = \sum\left(\frac{\Delta W}{g}\rho\omega^2\right)\cos\phi + \sum\left(\frac{\Delta W}{g}\rho\alpha\right)\sin\phi \quad [16]$$

and summing components of force parallel to the $y$-axis,

$$O = \sum\left(\frac{\Delta W}{g}\rho\alpha\right)\cos\phi - \sum\left(\frac{\Delta W}{g}\rho\omega^2\right)\sin\phi \quad [17]$$

The quantities $g$, $\alpha$, and $\omega^2$ are independent of the position of the particle $\Delta W$, hence Equations [16] and [17] may be written

$$\frac{Y}{K} = \frac{\omega^2}{g}\sum\rho\cos\phi\ \Delta W + \frac{\alpha}{g}\sum\rho\sin\phi\ \Delta W \quad [18]$$

and

$$O = \frac{\alpha}{g}\sum\rho\cos\phi\ \Delta W - \frac{\omega^2}{g}\sum\rho\sin\phi\ \Delta W \quad [19]$$

But, from Fig. 5,

$$\rho\cos\phi = x \quad [20]$$

hence

$$\sum\rho\cos\phi\ \Delta W = \sum x\ \Delta W = \int x\,dW \quad [21]$$

as $\Delta W$ approaches zero as a limit. This latter expression is recognized as the moment of the weight of the disk about the $y$-axis, which may be written

$$\int x\,dW = (Y + u)W \quad [22]$$

Also, from Fig. 5,

$$\rho\sin\phi = y \quad [23]$$

hence

$$\sum\rho\sin\phi\ \Delta W = \sum y\ \Delta W = \int y\,dW \quad [24]$$

as $\Delta W$ approaches zero as a limit. This expression is recognized as the moment of the weight of the disk about the $x$-axis, which may be written, from Fig. 5,

$$\int y\,dW = vW \quad [25]$$

Substituting [22] and [25] in [18] and [19],

$$\frac{Y}{K} = \frac{\omega^2}{g}(Y + u)W + \frac{\alpha}{g}vW \quad [26]$$

and

$$O = \frac{\alpha}{g}(Y + u)W - \frac{\omega^2}{g}vW \quad [27]$$

Solving Equations [26] and [27] for $Y$, from Equation [26]

$$Y = \frac{\omega^2 u + \alpha v}{\frac{g}{WK} - \omega^2} \quad [28]$$

and from Equation [27]

$$Y = \frac{\omega^2}{\alpha}v - u \quad [29]$$

In order to simplify the following equations the quantity $\sqrt{\frac{g}{WK}}$ will be indicated by the symbol $\omega_c$. Then

$$\frac{g}{WK} = \omega_c^2 \quad [30]$$

and Equation [28] becomes

$$Y = \frac{\omega^2 u + \alpha v}{\omega_c^2 - \omega^2} \quad [31]$$

But, from Equations [29] and [31],

$$Y = \frac{\omega^2}{\alpha}v - u = \frac{\omega^2 u + \alpha v}{\omega_c^2 - \omega^2} \quad [32]$$

and solving this for $u$,

$$u = \left[\frac{\omega^2}{\alpha} - \frac{\left(\frac{\omega^4}{\alpha} + \alpha\right)}{\omega_c^2}\right]v \quad [33]$$

In the above equation the coefficient of $v$ will be represented by the symbol $P$. Then

$$\left[\frac{\omega^2}{\alpha} - \frac{\left(\frac{\omega^4}{\alpha} + \alpha\right)}{\omega_c^2}\right] = P \quad [34]$$

whence

$$u = Pv \quad [35]$$

From the geometry of Fig. 5,

$$e^2 = u^2 + v^2 \quad [36]$$

where $e$ is the original eccentricity of the load as previously explained in the outline of Föppl's solution. Then from Equations [35] and [36]

$$e^2 = P^2v^2 + v^2 \quad [37]$$

and solving for $v$,

$$v = \frac{e}{\sqrt{1 + P^2}} \quad [38]$$

From Equation [35],

$$u = Pv = \frac{Pe}{\sqrt{1 + P^2}} \quad [39]$$

Substituting [38] and [39] in [29],

$$Y = \frac{\omega^2 e}{\alpha\sqrt{1 + P^2}} - \frac{Pe}{\sqrt{1 + P^2}} \quad [40]$$

or

$$Y = \frac{e(\omega^2 - \alpha P)}{\sqrt{\alpha^2 + (\alpha P)^2}} \quad [41]$$

and substituting for $P$ its value from Equation [34],

$$Y = e\,\frac{\omega^4 + \alpha^2}{\sqrt{\alpha^2\omega^4_c + [\omega^2(\omega^2_c - \omega^2) - \alpha^2]^2}} \quad [42]$$

This may be written

$$\frac{Y}{e} = \frac{\omega^4 + \alpha^2}{\sqrt{\alpha^2\omega^4_c + [\omega^2(\omega^2_c - \omega^2) - \alpha^2]^2}} \quad [43]$$

which is an expression for the coefficient of deflection, so called, an abstract number.

It may be of interest to investigate the variations in the angle $\theta$. From Fig. 5,

$$\tan\theta = \frac{v}{u} \quad [44]$$

But from Equation [39],

$$\frac{v}{u} = \frac{1}{P} \quad [45]$$

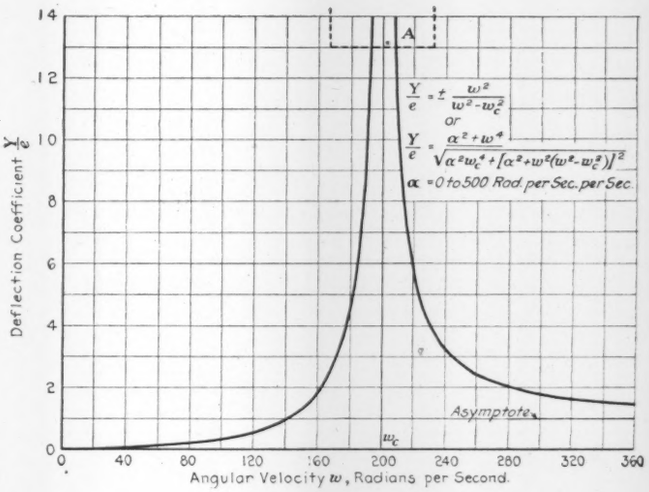


FIG. 6 RELATIONSHIP BETWEEN ANGULAR VELOCITY AND COEFFICIENT OF DEFLECTION

hence

$$\tan\theta = \frac{1}{P} \quad \ldots\ldots\ldots\ldots [46]$$

and substituting the value of $P$ from Equation [34], and simplifying,

$$\tan\theta = \frac{\omega_c^2\alpha}{\omega^2(\omega_c^2 - \omega^2) - \alpha^2} \quad \ldots\ldots\ldots\ldots [47]$$

Equations [43] and [47] have several interesting features. In plotting the following curves, a value of $\omega_c = \sqrt{\frac{g}{WK}} = 200$ radians per second was assumed as reasonable.

In Fig. 6 the coefficient of deflection $Y/e$ was plotted against values of angular velocity $\omega$ from Equation [43]. It is to be noted that as the speed becomes very large $Y/e$ approaches a value of unity; i.e., the shaft deflection approaches $e$ as a limit. Equation [47] shows that as $\omega$ becomes very great, $\tan\theta$ is negative and approaches zero as a limit. Thus $\theta$ approaches 180 deg. When $\theta$ = 180 deg. and $Y = e$, the disk is rotating about its center of mass, as can be seen from Fig. 5. Thus as the speed becomes very large the disk approaches a condition of rotation about its center of mass.

of accuracy in computing these ordinates, the substitution $\omega = \omega_c + \Delta\omega$ was made in Equation [43].

The curves of Fig. 7 show that the maximum shaft deflection occurs when $\omega = \omega_c = \sqrt{\frac{g}{WK}}$. Thus $\omega_c$ is the critical speed. It may be shown mathematically that the maximum shaft deflection does not occur exactly at $\omega = \omega_c$, but Fig. 7 shows that the difference is negligible. These curves also show that the maximum shaft deflection is decreased with increase in the acceleration. This explains why, in the operation of very high-speed turbines, the critical speed must be passed through rapidly to avoid dangerous shaft deflections.

Another interesting feature of Equation [43] is that when the substitution $\alpha = 0$ is made, this equation becomes

$$\frac{Y}{e} = \frac{\omega^2}{\pm(\omega_c^2 - \omega^2)} \quad \ldots\ldots\ldots\ldots [48]$$

which is Equation [10], Föppl's solution, except for the ambiguity of sign.

In Figs. 8 and 9 values of the angle $\theta$ were plotted as determined from Equation [47]. Fig. 8 shows the way in which the angle $\theta$

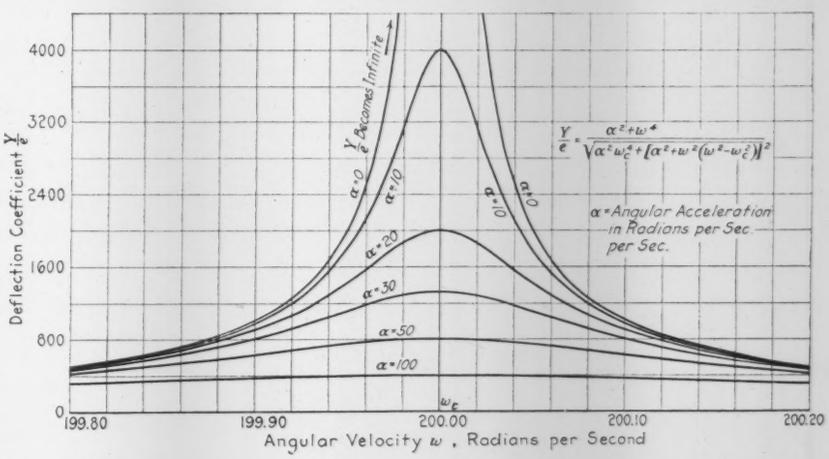
Deflection Coefficient Y/e
Angular Velocity ω, Radians per Second
Y/e Becomes Infinite
α = Angular Acceleration in Radians per Sec. per Sec.
ωc

FIG. 7 MAGNIFICATION OF REGION A OF FIG. 6, SHOWING THAT MAXIMUM SHAFT DEFLECTION OCCURS WHEN $\omega = \omega_c$

varies over the entire probable range of speeds, both increasing and decreasing. The center of gravity of the disk may be thought of as revolving about the center of the shaft at the disk (with respect to the $x$- and $y$-axes). It appears that, if the critical speed is passed

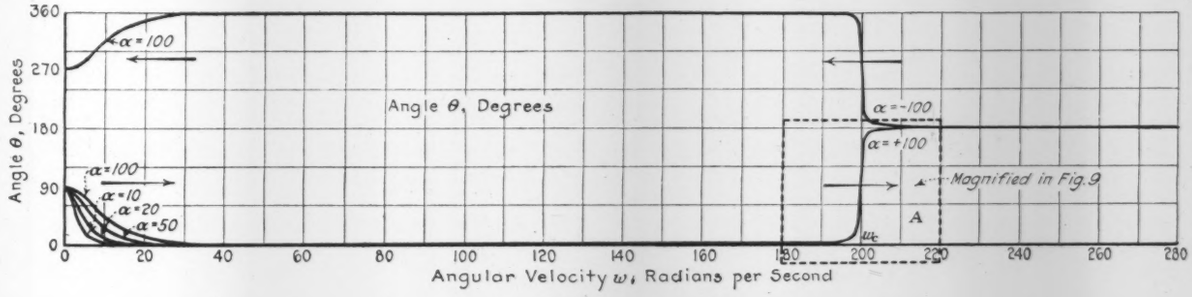


FIG. 8 VARIATION OF ANGLE $\theta$ OVER PROBABLE RANGE OF SPEEDS

It is found that for all speeds except those in the immediate proximity of $\omega = \omega_c$, the values of $Y/e$ from Equation [43] coincide almost exactly with the absolute values of $Y/e$ computed from Equation [10], $Y/e = \omega^2/(\omega_c^2 - \omega^2)$, the effect of acceleration being negligible. Thus Fig. 6 represents the curve corresponding to either Equation [43] or Equation [10], for any value of acceleration.

In order to investigate the region $A$ (Fig. 6) as $\omega$ passes through the value $\omega_c$, this region was magnified as shown by Fig. 7. Values of $Y/e$ were plotted against the angular velocity $\omega$, from Equation [43], for values of angular acceleration $\alpha$ of 100, 50, 30, 20, 10, and zero radians per second per second. In order to get a fair degree through with increasing speed, the center of gravity of the disk will apparently pass completely around the center of the shaft, and in the direction of rotation of the disk.

The curves of Fig. 8, for the critical region, are magnified in Fig. 9 and plotted for different values of acceleration. It may be of interest to note from these curves that the rate of change of the angle $\theta$ with *time*, at the critical speed, is approximately constant, being influenced but slightly by the value of the acceleration.

### MAXIMUM SHAFT DEFLECTION AT THE CRITICAL SPEED

The maximum shaft deflection, which occurs at the critical speed,

may be found by substituting $\omega = \omega_c$ in Equation [42], which then becomes

$$Y_{max} = e\frac{\omega^4_c + \alpha^2}{\sqrt{\alpha^2\omega^4_c + \alpha^4}} \quad [49]$$

This may be simplified to

$$Y_{max} = e\frac{\sqrt{\omega^4_c + \alpha^2}}{\alpha} \quad [50]$$

or

$$Y_{max} = e\sqrt{\left(\frac{\omega^2_c}{\alpha}\right)^2 + 1} \quad [51]$$

This is the shaft deflection *at the disk*.

It may be shown that for shaft and disk proportions ordinarily met with, the quantity $\left(\frac{\omega^2_c}{\alpha}\right)^2$ is very large as compared with unity. Hence the term 1 under the radical sign is negligible and Equation [51] may be written, approximately,

$$Y_{max} = e\frac{\omega^2_c}{\alpha} \quad [52]$$

But $\omega^2_c = \frac{g}{WK}$, hence

$$Y_{max} = \frac{e}{\alpha}\frac{g}{WK} \quad [53]$$

From Equation [11] the equivalent static load which, if applied to the shaft at the position of the disk, would produce this deflection, is

$$F = \frac{Y_{max}}{K} = \frac{e}{\alpha}\frac{g}{WK^2} \quad [54]$$

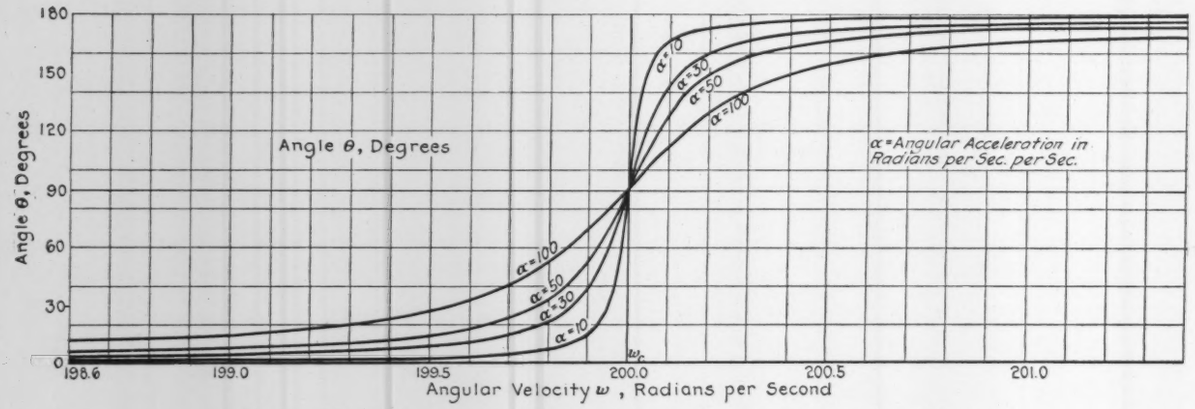


FIG. 9 MAGNIFICATION OF A PORTION OF THE CRITICAL REGION *A* IN FIG. 8

With a means of measuring the eccentricity of the center of gravity of the disk, the size of the shaft for any desired performance may be computed.

It can be shown that for any fixed eccentricity and acceleration there is one particular diameter of shaft in which the stress is a minimum.

# Industrial France

UNEMPLOYMENT, the cause of distress to the people of many countries, the relief of which has been encompassed with considerable economic difficulties, is practically non-existent in France. Indeed, the industry of the country has been able to absorb in activities thousands of immigrant workers. These —mainly of Italian, Polish, Belgian, or Spanish nationality— have settled there and, although the official figures do not include coal miners or agricultural workers, the two classes most in demand, the position may be gaged from the fact that the recorded figures for other workers reached during last year no less than 6000 immigrant workers in a week. This is not, however, the only feature of industrial interest in France today, for the industrial capacity of the country has been permanently increased. Alsatian ores, potash, and oil have added to the national resources. The manufacturing capabilities have also been greatly increased by the acquisition of the well-developed and varied engineering and textile industries of Alsace. Expansion has, however, been mainly caused through the reconstruction of almost the entire range of textile industries, of coal and iron mines, metallurgical and engineering works, glass and chemical works, in the devastated areas. In addition to this factor it is to be noted that industry was transplanted during the war from these areas to others more remote from the seat of operations and old factories were reëstablished and new ones built, which are all available and, indeed, are in use at present. France finds herself in the position of having her manufacturing industries completely modernized and equipped with new plant.

The machine-tool industry of the country has made great progress since the war, but is not yet able to meet the home requirements and orders must still be placed in America, Britain, and Germany. Increased activity may be anticipated, however, as a result of the growing capacity of France to produce special steels and the increased tendency towards specialization in production. Nothing very considerable has yet been done to obtain orders from abroad, but a central export agency has received an order from China of the value of $1^1/_2$ million francs. The French motor-car industry shows marked improvement due to the adoption of methods of mass production and to the condition of foreign exchanges. For the first three months of 1924 a total of motor vehicles of all classes of 16,700 tons was exported. This was more than double the figure for the same period in 1923. It is anticipated that increased trade with Great Britain will be obtained through the abolition of the McKenna duties. Aircraft construction is still on a small scale, but the creation of new air routes and the anticipated increase of use of aircraft in commercial and military operations has led to the laying down of new works.

Considerable progress has already been made in France in railway electrification and the Paris-Orleans, the Paris-Lyons-Mediterranean, and the Midi Railways have now got schemes in hand, involving in each case stretches of about 2000 miles of lines. These plans affect about one-third of the total French railway mileage. The building of electric locomotives has been pressed forward and a large works has been established at Tarbes which has sixty locomotives under construction at present for the Midi. The Paris-Orleans Railway have under order 80 electric locomotives which are to be built by the Société d'Etudes pour l'Electrification des Chemins de Fer Français, and construction has been started on part of an order for 200 electric locomotives, of which 80 are to be of the Oerlikon type of 1720 hp. Five of these are to be built at the Oerlikon works near Zurich, and the remainder at the French works at Ornans in connection with the Société de Construction des Batignolles and the Compagnie Générale de Construction des Locomotives. (*Engineering*, Sept. 5, 1924, pp. 332, 333.)

# Wind Power for Farm Electric Plants

## Test Data on a 15-ft. Wheel—Losses Due to Air Drag—Best Size of Wheel for Farm Electric-Light Plants—Combination Plant for Generating Electric Power and Pumping

By F. J. PANCRATZ,[1] CHICAGO, ILL.

MANY experiments are being conducted to develop a practical and commercial windmill that will supply the average farmer with power to generate the necessary electricity for lighting and pumping, and for operating the many appliances that are used on a modern farm. At present the windmill is used extensively for pumping water. Gasoline-engine lighting plants have proved popular but are expensive in maintenance, and there is no plant that is automatic in operation. The power companies find that running pole lines to isolated farms is too expensive, and as a consequence the agricultural districts offer a big market for a means for supplying the necessary electric power that is at once practical, cheap, low in maintenance cost, and reliable.

According to the U. S. Weather Bureau and tests, the wind distribution is favorable throughout the agricultural district, with a few exceptions, for deriving power from this source. Periods of calm occur in cycles, with the lowest average wind velocities during the summer months. The cycles of wind velocities are completed during periods varying from 5 to 13 days in length, depending upon the season of the year. The longest calm or period of continuous low wind below five miles per hour is 34 hours for the Middle West, and varies with the locality. The windmill must be efficient in low winds in order to minimize the storage-battery capacity. Power can be generated in all winds above 5 miles per hour and water pumped direct in winds above 8 miles per hour with a comparatively small wheel. However, the present type of wheel is not satisfactory for such operation.

### Results Obtained in Tests of a 15-ft. Windmill

A test of a 15-ft.-outside-diameter wheel carried on for over a year gave many interesting results and showed the defects and the many conditions that had to be met to make it a commercial success. The data collected during the period of this test are very instructive and can be readily used to check the theories, especially those derived in the development of the airplane and which are applicable to the windmill. Aeronautical engineers have made great progress. Their data are reliable, checked and rechecked by machines in actual practice. These data and the results of the tests on windmills are a great aid in designing a windmill that will generate the maximum power with the least vane surface, meet all weather conditions, and compete under the present market conditions.

A windmill having a 15-ft. outside and an 8-ft. inside diameter and 32 equally spaced vanes was mounted upon a 60-ft. tower in a suburb of Chicago. This wheel was held into all winds by a fixed tail. The north and south exposure was very good. A wooded ridge running north and south about 4000 ft. east of the windmill and having a gradual slope and elevation of 10 ft. below the center of the wheel, obstructed the east and west winds to some extent. This wheel ran in all storms. It withstood all weather conditions for over one year, and generated electric power for nearly six months during the summer and fall. The generator caused some trouble at times, so the test was not continuous. Eleven days were lost in necessary repairs and to change the generator's characteristic curve. This generator charged a 150-ampere-hr. 16-cell battery. The circuit was controlled by a "cut out" switch to prevent reversal of current from the battery to the generator. A reversal of $1/4$ ampere was sufficient to open the switch, so the loss of power due to the generator's acting as a motor was very low. Thirty-two-volt lamps were used to consume the stored energy. An anemometer was installed near the wheel, but far enough away so as not to be affected by the changes in velocities about the wheel. This instrument was carefully maintained and checked. By means of a recording graphic chart, a continuous record of wind velocities was obtained. A graphic recording ammeter was connected in series between the battery and generator. A record was also kept of the charging and discharging of the battery, battery density, and number of ampere-hours consumed by the bank of lamps.

A standard make of 32-volt 25-ampere 1800-r.p.m. ball-bearing generator was reconstructed and a shifting third brush added to suit the required characteristics of the wheel. It was mounted at the base of the tower for convenience in testing and mechanically connected to the wheel through two sets of reduction gears, a long vertical shaft, and one set of change gears. By means of the change gears and the shifting of the third brush, the best loading of the wheel was easily obtained for any wind velocity. Before mounting, the generator was carefully tested for all the losses and its speed-ampere output curve determined for a wide range of speeds and brush settings. All shafts were mounted on ball bearings. The

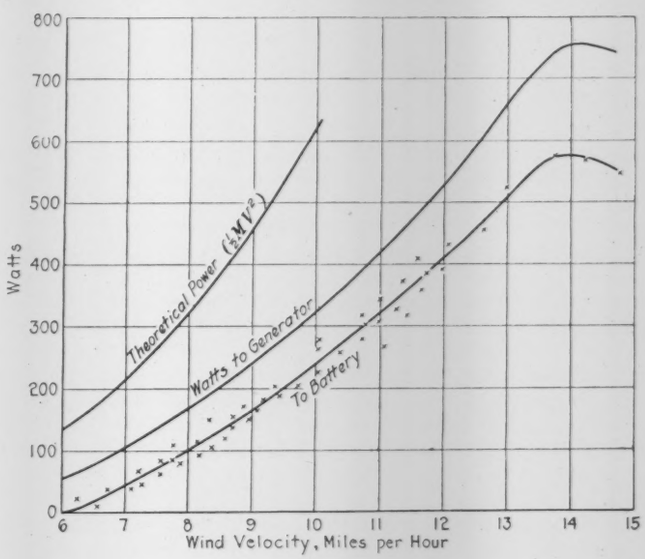


Fig. 1

power developed by the wheel was determined by the output of the generator plus the estimated losses in the gears and bearings. Fig. 1 shows the charging rates at different wind velocities for the 15-ft. wheel with the best brush setting and gear ratio. It will be noted that many readings were not obtained near the peak of 18 amperes. This was due to the wheel's racing ahead of its best speed on decreasing load beyond the peak, and because the wind was not steady. The theoretical available power is based upon the formula $\frac{1}{2} MV^2$, where $M$ is the mass of air intercepted between two circles of radii $7\frac{1}{2}$ ft. and 4 ft., respectively, with air moving at the free-air velocity $V$. The readings were taken during the month of August, 1920.

The maximum r.p.m. of wheel recorded was 210. This was obtained with the "cut out" switch blocked and corresponds to a charging rate of $-2$ amperes for the brush setting and gear ratio used. Wheel speeds 150 to 190 r.p.m. were common, but these are too high for good operation and long life. The generation "cut in" at 1000 r.p.m. in a six-mile wind, so the generator speeds were excessive.

The amount of power delivered to the battery for five months is given in kilowatt-hours in Table 1. The average per day are taken from the total for each month. The wind velocities in the last column are those of the U. S. Weather Bureau of Chicago and are included for comparison.

During this test 44 hr. was the longest period of calm below a 6-

[1] McAlear Manufacturing Co. Assoc-Mem. A.S.M.E.

Presented at the Spring Meeting, Cleveland, Ohio, May 26 to 29, 1924, of The American Society of Mechanical Engineers. All papers are subject to revision.

mile wind. On an average the battery was overcharged three times a month. The amount of power wasted depends upon the size of battery and generator, wind distribution, battery efficiency, and character of load. Considering these factors with the load

TABLE 1 DATA ON TEST OF 15-FT. WINDMILL

| Month during year 1920 | Kw-hr. charged for month | Kw-hr. charged, average per day | Wind vel., average per day | Wind vel., average per day for 10 yr. |
|---|---|---|---|---|
| June | 72.4 | 2.4 | 11.1 | 10.9 |
| July | 81.0 | 2.6 | 9.6 | 9.7 |
| August | 62.3 | 2.0 | 8.5 | 10.3 |
| September | 69.0 | 2.3 | 11.6 | 11.4 |
| October | 65.2 | 2.1 | 11.0 | 12.1 |

required on a modern farm, the generator was too large and the 150-ampere-hr. battery was ample to carry over any calm. A battery of the above capacity allows a discharge rate of 3.4 amperes

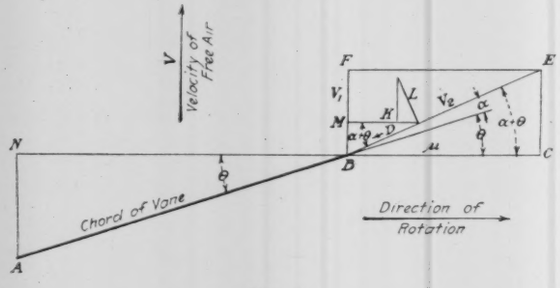


FIG. 2

per hour continuously for the 44 hr. of maximum length of calm. During the summer months the farmer's lighting load is lower than in the winter months. It is estimated from the relative wind velocities and the results of the test that an average of 4.4 kw-hr. can be delivered during the winter months.

The results of the test are favorable for output, but the construction used is too costly, this being due to the large amount of vane surface necessary to generate the required power, and to the high speeds involved. It is important that the wheel should deliver electric power in lower winds. If this test wheel had begun charging in a 5-mile wind much power would have been gained, besides a smaller battery could have been used. The high speed of a wheel with sharp vane angles can be slowed up by means of a special curved vane without reducing the efficiency in low winds. The percentage of the wheel's projected area to be covered and the shape of the vanes are important factors. The generator capacity plus the losses should suit the output of the wheel and the generator should have very low field, copper, and iron losses, with the maximum efficiency at approximately one-third full load. In the case of a high-speed wheel the losses due to air drag by the structural member or braces of the vanes become a more important factor in the design of an efficient wheel.

## LOSSES DUE TO AIR DRAG

In order to make a careful study of the foregoing factor it is necessary to develop and apply the theory of air movement against vanes. Many wind-tunnel tests give accurate data for lift and drag on the various shapes of wing sections used in airplanes, and these can be used to a good advantage in designing the best vane for any particular size of wheel. The results obtained in the test previously mentioned serve as a check upon the theories and wind-tunnel data.

Referring to Fig. 2, $AB$ is the chord of an elementary strip of vane, which is assumed to be not isolated from the neighboring elements.

Let $V$ = velocity of the free air in miles per hour
$V_1$ = average velocity of air through the wheel
$A$ = projected area of chord (= $NB \times$ width) in square feet
$L$ & $D$ = lift and drag coefficients, respectively, their values depending upon the aspect and shape of the curve having $AB$ as chord
$\alpha$ = angle of incidence
$V_2$ = velocity in miles per hour relative to vane.

Then, resolving the forces $L$ and $D$ in the plane of the wheel into $L \sin(\alpha+\theta)$ and $-D\cos(\alpha+\theta)$ (in lb. per sq. ft. per mile wind) and taking $A/\cos\theta$ as the projected area of the chord,

$$\text{Running pressure} = [L\sin(\alpha+\theta) - D\cos(\alpha+\theta)]\, A/\cos\theta\, V_2^2$$

But $V_2^2 = V_1^2/\sin^2(\alpha+\theta)$; therefore, substituting in the preceding equation,

$$\text{Running pressure} = [L\sin(\alpha+\theta) - D\cos(\alpha+\theta)]\,\frac{AV_1^2}{\cos\theta\sin^2(\alpha+\theta)}$$

Now $u$ = speed of section $AB$ in miles per hour = $V_1\cot(\alpha+\theta)$, and running pressure $\times\, u$ = power; consequently, multiplying the running pressure by the value of $u$ and reducing in terms of watts, the equation becomes—

$$\text{Watts} = 1.99\,C\,[L\sin(\alpha+\theta) - D\cos(\alpha+\theta)]\,\frac{AV_1^3\cot(\alpha+\theta)}{\cos\theta\sin^2(\alpha+\theta)} \quad [1]$$

$C = 1$, provided no interference takes place before entering and after leaving the strip $AB$.

The summation of the watts of all the elementary strips with their respective values of $V_1$ and $\theta$ gives the total power of the vane. Strips 0.5 ft. wide with the average corresponding values of $V_1$ and $\theta$ give a sufficiently accurate value in watts. If the angle of incidence is the same for all strips, the values of $L$ and $D$ will remain constant for similar sections.

The value of $u$ varies as the radius, so at any radius $r_1$, $\cot(\alpha+\theta)$ varies as $r_1/R$ when $V_1$ is constant, where $R$ = outer radius of vane. This determines the relation of $\theta$ and $V_1$ at any radius $r_1$.

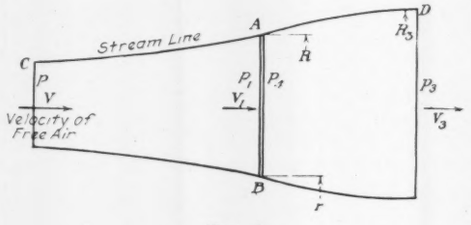


FIG. 3

This value of $\theta$ is assumed at the outer radius of vane, and by trial the best angle is determined for different values of $\alpha$, the angle of incidence. The best type of vane and the amount of warping can be readily arrived at for non-interference conditions.

The relation between $V_1$ and $V$ must be determined. We know $V_1$ is not uniform at all radii when passing through a wheel. The average value can be found by means of the theorem that the velocity head plus the static head equals a constant.

Referring to Fig. 3, $R$ and $r$ are respectively the outer and inner radii of vane $AB$.

$$M = \frac{w}{g}$$

$$\text{Total energy} = \frac{1}{2}MV^2 \text{ (in ft-lb. per sec. units)}$$

and

$$E = 0.00366\,(R^2 - r^2)\,V^3$$

Here $w$ = 0.075 lb. per cu. ft. and $g$ = 32.2

$E_1$ = energy of air passing between circles of radii $R$ and $r$

$$= 0.00366\,(R^2 - r^2)\,V_1\,(V^2 - V_3^2)$$

The area of air section at $C$, or in free air that passes through the wheel, is

$$\frac{V_1}{V}\pi\,(R^2 - r^2)$$

therefore $\pi(R^2 - r^2) - \pi\,\frac{V_1}{V}\,(R^2 - r^2)$ = area of air section in

front of wheel at $C$ that passes around the wheel at $R$ and $r$. The pressure perpendicular to $AB$ due to this air is

$$F = K \frac{V - V_1}{V} \pi (R^2 - r^2) V^2$$

$K$ is a constant depending upon the aspect value and when $R/r = 1.8$, $K = 0.00172$. The energy of the air passing around $AB$ transmitted to the wheel is $FV_1$, therefore

$$E_2 = 0.0054 (V^2V_1 - VV_1^2)(R^2 - r^2)$$

The total energy available to wheel is

$$E_1 + E_2 = 0.00366 (E^2 - r^2) (V_1V^2 - V_1V_3^2 + 1.47V^2V_1 - 1.47 VV_1^2) \ldots [2]$$

This equation has two unknown quantities, $V_1$ and $V_3$. $p$, $p_1$, $p_4$ and $p_3$ are pressures at the sections shown and their respective velocities are $V$, $V_1$, and $V_3$. Velocity head plus pressure head equals a constant,

$$p + \tfrac{1}{2} dV^2 = p_1 + \tfrac{1}{2} dV_1^2$$

and

$$p + \tfrac{1}{2} dV_3^2 = p_4 + \tfrac{1}{2} dV_1^2$$

where $d$ = density $\div$ $g$. The density of the air is assumed constant. By subtraction

$$p_1 - p_4 = \tfrac{1}{2} d [V^2 - V_3^2]$$

or the difference of pressure per unit area, whence

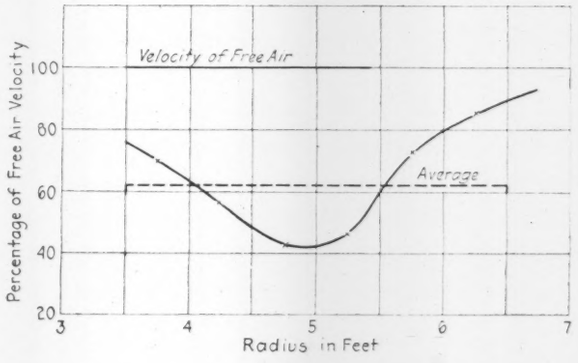


FIG. 4

$$P \text{ (total pressure)} = dAV_1 (V - V_3)$$

= mass of air through wheel times change in velocity

and

$$\frac{P}{A} = dV_1 (V - V_3) = \text{pressure per unit area}$$

therefore

$$dV_1 (V - V_3) = \tfrac{1}{2} d (V^2 - V_3^2)$$

Dividing both members by $d$ $(V - V_3)$,

$$V_1 = \tfrac{1}{2} (V + V_3)$$

Substituting this value of $V_1$ in Equation [2], differentiating, placing the first derivative equal to zero, and solving for value of $V$,

$$V = 4.18 V_2, \quad \therefore V_3 = 0.24 V$$

But

$$V_1 = \frac{V + V_3}{2}, \quad \therefore V_1 = 0.62V$$

These average values of $V_1$ and $V_3$ in terms of $V$ are conditions for maximum power to wheel with $7^1/_2$-ft. outer and 4-ft. inner radius. Substituting these values in the Equation [2],

$$E_1 + E_2 = 0.0034 (R^2 - r^2) V^3$$

$\frac{E_1 + E_2}{E}$ = 93 per cent of the total energy available to the vanes of the wheel. The nearer unity the value of $R/r$ becomes the higher the value of $K$, consequently the greater the percentage available. Fig. 4 shows the value of $V_1$ in terms of $V$ at different radii for a wheel with a $6^1/_2$-ft. outer and a $3^1/_2$-ft. inner radius.

In the power Equation [1] the factor $C$ depends upon the interferences. In a practical wheel the vane surface must be kept reasonably near the center of rotation in order to have simplicity in construction. This necessarily increases the interference and $C$ becomes less than unity, therefore the vane surface cannot develop the maximum power.

Interference is a big factor in checking the speed of a wheel in high winds. In Fig. 5 let

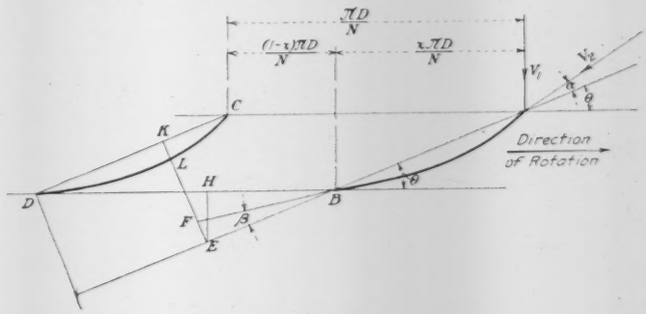


FIG. 5

$x$ = percentage of the circumference covered by the projected section of vane at a diameter $D$

$N$ = number of vanes

$\frac{x\pi D}{N}$ = area of the projected vane chord, assuming a strip 1 ft. in width

$E = 0.0034 (R^2 - r^2) V^3$

The average difference of pressure per sq. ft., taking $V_1$(avg.) = 0.62 $V$, is

$$P_1 = 0.00175 V^2 \text{ (ft-sec. units)}$$

The mass of air per vane 1 ft. wide and having an average diameter of $D$ is

$$M = 0.004 \frac{DV}{N}$$

$BE$ is the distance along chord between leaving tip and point of maximum ordinate of adjacent vane at a diameter $D$, or

$$BE = \frac{\pi D}{N} \cos \theta - \frac{5}{8} \frac{\pi Dx}{N \cos \theta} \ldots\ldots\ldots\ldots [3]$$

the maximum ordinate being taken at $^3/_8$ from the leading tip.

Also

$$HB = BE \cos \theta$$

so

$$HB = \frac{\pi D}{N} \left( \cos \theta - \frac{5}{8} \frac{x}{\cos \theta} \right) \cos \theta$$

The pressure for the length $HB$, assuming strip to be 1 ft. wide, is

$$P = 0.00175 V^2 \frac{\pi D}{N} \left( \cos^2 \theta - \frac{5}{8} x \right)$$

This pressure must shift the total mass of air per vane which passes through the wheel the distance $KL + FE$ during the time which is required to travel from $B$ to $E$, in order that there shall be no interference.

$V_2$ = velocity along vane

$$= \frac{0.62\,V}{\sin(\alpha + \theta)}$$

and

$$\frac{BE}{V_2} = t = \text{time required to travel from } B \text{ to } E.$$

Substituting these values in Equation [3],

$$t = \frac{5.07\,D}{NV}\left(\cos\theta - \frac{5}{8}\frac{x}{\cos\theta}\right)\sin(\alpha + \theta) \quad \ldots\ldots [4]$$

The acceleration perpendicular to the chord $DC$ is

$$\frac{P}{M}\cos\theta = 0.75\,V\left(\cos^3\theta - \frac{5}{8}x\cos\theta\right)$$

For no interference the distance

$$\tfrac{1}{2}\left(\frac{P}{M}\cos\theta\right)t^2 \geq KL + FE$$

Substituting the value of $t$ from [4],

$$KL + FE \geq \frac{3.57\,D}{VN^2}\left(\cos^3\theta - \frac{5}{8}x\cos\theta\right)\sin^2(\alpha + \theta)$$

$KL$ is the maximum ordinate of the vane at $^3/_8$ the distance from the leading tip and $FE$ has been determined in wind-tunnel tests for different-shaped curves, so that value of $x$, the percentage of covering, can be determined for any wind velocity. For a circular-section vane the factor $^5/_8\,x\cos\theta$ would change to $1/2\,x\cos\theta$. For no interference the percentage of covering diminishes when the angle decreases or the wind velocity increases. For a given vane the interferences increase when the wind velocity increases above a certain value. This accounts for the dropping curve of wheel speed and wind velocity. In higher winds this interference effect can be increased more rapidly which would give a means of speed control and solve the high-speed problem of wheels running in storms.

Fig. 6 shows the curve to be compounded on the pressure side of vane in order to increase the interference and drag in winds above a predetermined velocity and thereby prevent higher peripheral velocity when in stronger winds without decreasing the efficiency of the wheel in winds below that velocity.

$V_2$ = velocity in ft. per sec. relative to vane. (The velocity along the vane is assumed to be the same as the entry velocity, which is sufficiently accurate in this case.)

$y$ = deflection, perpendicular to $AD$, of air along the vane at any point $C$

Let $d$ = distance of $C$ along $AD$, weight of air taken = 0.075 lb. per cu. ft., and $g = 32.2$. Then the mass of air which is influenced by the deflection of air next to the curve is $M = 0.0163\,V_2y$. The average deflection of this mass is $^1/_2\,y$, and $d/V_2$ is the time required for the air to travel from $B$ to $C$. If the air follows curve $BC$, this will be the time required to deflect the air next to the vane the distance $y$.

$F$ = force perpendicular to $AD$ due to deflection of air. The area of vane effecting this deflection for a strip 1 ft. wide is $d$,

$$\therefore F = K_1 d V_2^2$$

where $K_1$ is a constant or suction factor whose value depends upon the rate of change in curvature. This force produces an acceleration $a$ perpendicular to $AD$, and

$$a = \frac{F}{M}$$

Substituting the values of $F$ and $M$ in this equation,

$$a = \frac{61.3\,KdV_2}{y}$$

Also

$$\tfrac{1}{2}\,y = \tfrac{1}{2}\,a\,t^2$$

where $t$ = time of deflection. Substituting values of $a$ and $t$ in this equation,

$$y^2 = \frac{61.3\,Kd^3}{v_2} \quad \ldots\ldots [5]$$

But $V_2 = \dfrac{V_1}{\sin(\alpha + \theta)}$, and taking the average value of $V_1 = 0.62V$, which depends upon the aspect of vanes and wheel, and substituting in [5],

$$y^2 = \frac{38.0\,Kd^3\sin(\alpha + \theta)}{V} \quad \ldots\ldots [6]$$

This is the equation of the stream-line curve. The air moving at the velocity $V$ (ft. per sec.) when passing through the wheel, which latter is traveling at a definite peripheral speed determined by the angle of incidence, vane angle, and the air velocity through the wheel, will follow the curve of Fig. 6 without loss. The increase in skin loss is negligible. The arrows indicate the exchange of pressure along the curve. In a stream line these balance or become equal to zero along $AD$, consequently no energy is lost. When the velocity along the vane increases the air "breaks" before reaching

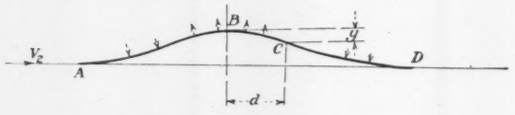


FIG. 6

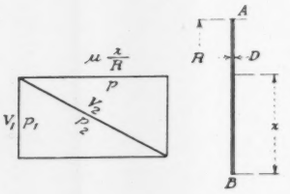


FIG. 7

and after passing $B$, causing a drag in the direction of air travel along the vane. In addition, this deflection of air increases the interference with the air on the suction side of the adjacent following vane. This increase in drag and interference will reduce relative wheel speed in higher winds. The interference is the most important factor.

By means of the above-mentioned curve a sharp vane or small value of $\theta$ can be used and the r.p.m. of wheel held to a comparatively low figure. The compound curve will give the best results on a vane having a small value of $\theta$. Besides, the vanes will develop greater power per square foot of surface in lower winds.

The relatively high velocity of the wheel during periods of low winds will necessitate in arranging the structural members or braces of vanes for a minimum loss in order to have an efficient wheel. These members must have sufficient strength and stiffness to withstand the highest wind velocities.

The loss or air drag can be determined for standard members. Referring to Fig. 7, a member revolves about $B$ as an axis. The projected area of a differential strip at a radius $x$ is $Ddx$, where $D$ = diameter or width in feet. Let $p_2$ = pressure per unit area; then

$$p_2 = K_2 V_2^2 D dx$$

where $K_2$ is the resistance coefficient, which is not absolutely constant but is a function of $V_2D$. Resolving this pressure in the plane of rotation,

$$p = K_2 D V_2 u \frac{x}{R}\,dx \quad \ldots\ldots [7]$$

where $u$ = velocity at radius $R = 2\pi Rn$ and

$n$ = revolutions per second.

Substituting these values in [7],

$$p = 2\pi K_2 D n V_2 x dx \quad \ldots\ldots [8]$$

But

$$V_2 = \sqrt{V_1^2 + (2\pi nx)^2}$$

Substituting this value in [8],

$$p = 2\pi K_2 Dn \sqrt{V_1^2 + (2\pi nx)^2}\, xdx$$

The power loss will then be

$$pu \frac{x}{R} = 4\pi^2 K_2 Dn^2 \sqrt{V_1^2 + (2\pi nx)^2} x^2 dx$$

Integrating between limits of $R$ and 0 and expressing the resulting equation in terms of watts,

$$\text{Watts loss} = 84.0\, K_2 n^3 DR \left[\left(R^2 + 0.0127 \frac{V_1^2}{n^2}\right) \sqrt{R^2 + 0.0254 \frac{V^2}{n^2}}\right] \ldots [9]$$

In which the factors are in foot-second units. The second term of the second member of this equation has been omitted, as it is a negligible quantity. The skin-friction loss of air flowing parallel with surfaces can be determined from resistances given by Dr. Zahm's experiments.

The application of the power formula for one vane in free air with no interference gives very interesting results as shown in the curves of Fig. 8. The watts developed are for different vane angles and angles of incidence of a standard vane. There is not much variation of power between angles of incidence of 4 to 8 deg.

### BEST SIZE OF WHEEL FOR FARM ELECTRIC-LIGHT PLANTS

The calculations have been carried through for several sizes of wheels with different types of vanes, different vane angles, and different angles of incidence, with their respective best coverings.

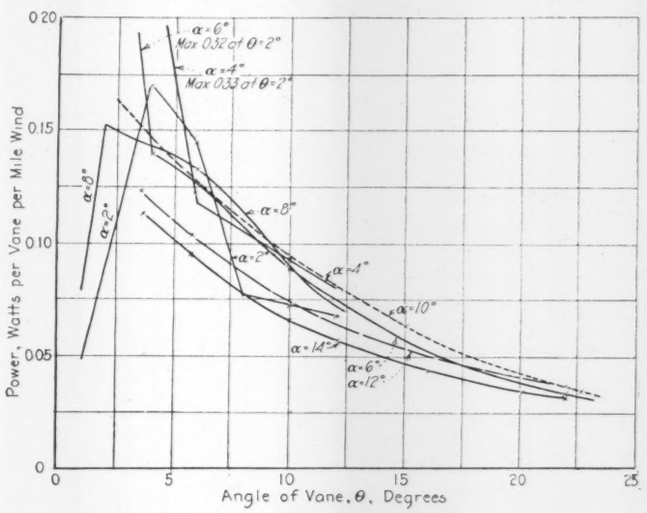


FIG. 8

A wheel 13 ft. in outside diameter and 7 ft. in inside diameter will give ample power for an electric plant suitable for a modern farm in the Middle West. A vane angle of 12 deg. at the outer tip is about the minimum that can be used in order that the wheel shall start in a sufficiently low wind. The losses of the necessary structural members increase nearly as the cube of the speed of the wheel, so some vane angle with a given construction of wheel will develop the maximum power. From the power formula and data of values of lift and drag, the most efficient type of vanes can be readily chosen. In Table 2 the vane is divided into six equal parts and the torque, power, etc. calculated for the average radius of each section, for one vane and a free-air velocity of one mile per hour. The values of $V_1$ are taken from Fig. 4 and are the true velocities through the wheel. The area of the vane is 3.0 sq. ft.

The r.p.m. of the above vane is 6.75 per mile wind (free air) when $\alpha = 4$ deg. In an 8-mile wind, interference at the outer radius begins when approximately 6 per cent of the projected circumference is covered. In this wind 93 per cent of 242 watts = 225 watts is available to vanes. Each vane with no interference consumes 22.8 watts, so 225/22.8 = 9.8 vanes are required. But due to interference this number must be increased. The correct covering for an 8-mile wind at the outer circumference is approximately 29 per cent. This will give the maximum power with the least square feet of vane surface. Twelve vanes of 1 ft. length of chord will give the necessary covering. In order to have the necessary strength at the inner radius of vane, the vane is given the same width throughout its entire length. The percentage of covering at the inner radius will be 54 per cent. Correction is made for the increase of interference. Ninety-three per cent of 59 watts = 54.8 watts available to this size of wheel in a 5-mile wind. The brace members of the wheel cause a loss of 5.0 watts running at 33.8 r.p.m. Assuming the gear and bearing efficiency to be 96 per cent, the vane efficiency in this wind velocity is 68 per cent. This leaves 31 watts to be delivered to the generator. According to the wind distribution during the summer months in the Middle West, a 36-volt, 12-ampere, 1100-r.p.m. at "cut in," third-brush generator is best suited for this size of wheel. The generator should have its maximum efficiency of about four amperes in order to have a low field loss. The idle loss for this generator on ball bearings can be kept below 29 watts. Consequently the wheel supplies sufficient power to begin charging in a 5-mile wind. The best gear rate is 32.5, requiring a back-gear shaft mounted on ball bearings. The third-brush non-sparking generator can be mounted next to the wheel on the tower. It will require attention only once a year. The maximum output of 12 amperes will occur in an 11-mile wind, and the generator will run at 2400 r.p.m. This corresponds to a wheel velocity of 75 r.p.m.

TABLE 2

| Radius in feet | $V_1$, velocity through wheel | Running pressure lb. | Torque in ft-lb. | Watts to vane | Watts delivered by vane | Efficiency of section, per cent | Per cent of total watts delivered by section | Per cent of total torque delivered by section |
|---|---|---|---|---|---|---|---|---|
| 3¾ | 0.70 | 0.0010 | 0.0037 | 0.0045 | 0.0036 | 81 | 12 | 12 |
| 4¼ | 0.57 | 0.0008 | 0.0034 | 0.0046 | 0.0034 | 74 | 11 | 9 |
| 4¾ | 0.42 | 0.00054 | 0.0026 | 0.0047 | 0.0025 | 53 | 8 | 8 |
| 5¼ | 0.46 | 0.00067 | 0.0035 | 0.0057 | 0.0034 | 60 | 11 | 11 |
| 5¾ | 0.73 | 0.00137 | 0.0079 | 0.0104 | 0.0076 | 73 | 25 | 25 |
| 6¼ | 0.85 | 0.00177 | 0.0111 | 0.0147 | 0.0106 | 72 | 33 | 35 |
| Avg. | 0.62 | Total | 0.0322 | 0.0446 | 0.0311 | 70 = Avg. | | |

The compound curved vane can be so proportioned that the interferences between vanes will increase very rapidly at slightly below 75 r.p.m. This will prevent the generator in running "over the peak," so that in all winds above 11 miles per hour it will generate near its maximum output of 12 amperes.

In order to develop the maximum power throughout the range of wind velocities between 5 and 11 miles per hour, the generator at the corresponding speeds must give the proper full loadings for the wheel. According to the curves in Fig. 8, the angle of incidence may vary from 4 to 10 deg. for $\theta$ above 12 deg. with not much change in output capacity of wheel. This corresponds to a variation of 34 per cent in wheel speed. The characteristic ampere-speed curve of a third-brush generator can readily be such that the wheel will "grab" its maximum loadings up to the 11-mile wind.

The maximum charging rate of 12 amperes is low for a 150 ampere-hr. battery. This is favorable for a long life for the battery, and besides gives better voltage regulation for lamp service. Overcharging at a 12-ampere rate will not injure this size of battery. Intermittent charging and discharging will increase the capacity of the battery to approximately 210 ampere-hr. The longest calm below a 5-mile wind seldom exceeds 34 hr. If the battery was completely charged, this would allow 197 watts average load per hour, or 788 watts for one-quarter of the time or six hours per day. Periods of higher wind velocity and calms occur in cycles, and these periods of higher winds are of sufficient duration to completely charge the battery. Assuming that the average charging rate is 10 amperes during the higher-winds velocity period, the battery will be charged in 21 hr. A simple "cut out" switch and the fact that the wheel can be left running in any wind without injury make the operation entirely automatic.

The twelve compound-curved vanes having an area of 36.0 sq. ft. allows a simple construction of wheel that will withstand all

storms and maintain a speed of an approximate maximum of 75 r.p.m. in winds above 11 miles per hour. The centrifugal forces in the wheel will be comparatively low. The modulus of section of this type of vane is ample to withstand the combined stresses of wind pressure and the centrifugal force due to rotation. A wide vane at the inner radius or rim permits a simple connection and gives sufficient stiffness to the vane so that it will hold its relative angular positions. The comparatively low velocities of wheel and generator will give quiet operation during periods of storm. The larger open center of wheel results in steadier running of the wheel and prevents its surging about the vertical axis of the tower. This was shown by the test wheel in operation.

The 36.0 sq. ft. of vane surface gives a comparatively small wheel.

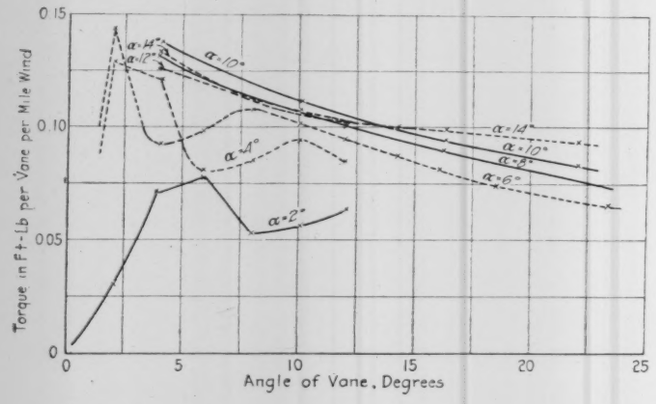


FIG. 9

The ordinary 10-ft. steel windmill used for pumping has over 50 sq. ft. of vane surface. The common 8-ft. pump mill has over 32 sq. ft. of vane surface. The ordinary tower for a 10-ft. wheel has sufficient strength to carry this wheel through all storms. A steady-running wheel with fixed tail requires less strength in a tower than the present closing type of mills.

By comparison with results of test and estimates, the 13-ft. wheel with 32 sq. ft. of vane surface would have charged 2.8 kw-hr. average per day instead 2.0 kw-hr. (results of tests for the month of August, 1920), which is an increase of 40 per cent. Besides it was able to charge in a lower wind, thus reducing the length of calm and in consequence the size of battery. It is difficult to calculate what the increase would be during the winter months, but since in stronger winds than 11 miles per hour the generator will give near its maximum output, the increase should be greater.

### COMBINATION PLANTS FOR GENERATING ELECTRIC POWER AND PUMPING

A combination of generating electric power and pumping water direct is very desirable in many localities. This is feasible. Fig. 9 shows that the running torque increases rapidly with the smaller vane angles. The small-angled vane would not be able to start a pump load from a standing position. The electric generator offers a means to "throw in" a pump such as is used in the agricultural districts, at a predetermined amperage or wind velocity. In an 8-mile wind the 13-ft., 12-vane-12-deg.-vane angle wheel produces a running torque of 148 ft-lb. at 54 r.p.m. and a maximum torque of 183 ft-lb. at 39 r.p.m. Connecting the pump load will reduce the wheel speed for a given wind velocity and the output of generator will be lowered. The pump load will produce a pulsating charging current. This will occur once per stroke in a single-acting pump. The flywheel effect of the wheel at 54 r.p.m. and generator at 1750 r.p.m. will be considerable, so that pulsations will be reduced in amplitude. The combined loads will work together satisfactorily, but the wheel will not develop its maximum power while pumping. The common-size pump ordinarily used with the present windmills does not require much power but the wheel must develop the maximum torque required at the center of the pump stroke. The wheel in question has sufficient torque to drive the generator at the reduced speed and the size of pump used with the present 8-ft. windmill. A manually operated single-pole switch controls the clutch circuit and whenever the wind "comes up" to, say, 10 miles per hour, it is thrown in automatically. The wind velocity varies continuously, so by connecting the pump load when the wind velocity is above 8 miles per hour excessive surging of wheel velocity is prevented. The clutch disconnects when the wind velocity decreases to 8 miles per hour. So pumping will not interfere with charging in a lower wind and the wheel will run under the best loading. The range of pump strokes will be from 16 to 30 per minute. This range is low due to the speed control of the special vane used. The strokes per minute can be changed to suit the pump, but the ratio of minimum to maximum strokes per minute will hold for this size of wheel and generator. The clutch, reducing bevel gears, and pump rod from the top of the tower are simple in construction and do not increase the friction losses for charging when the pump is disconnected. When the manually operated switch is closed it will be automatic in operation. After sufficient water is pumped the switch is opened and the mill only charges the battery.

The characteristics of a windmill are not favorable for large units. A 20-ft. wheel is about the largest size that can be used and give satisfactory results for generating electric power. The smaller-size wheels offer a cheap and reliable means of supplying the average farmer or isolated places with the necessary electric power and water. The first cost, cost of operating, and depreciation compare very favorably with those of the gasoline lighting plants.

## Discussion

E. N. FALES,[1] in his discussion of Mr. Pancratz's paper, dealt with the maximum power obtainable from a windmill. He pointed out that two German investigators, Dr. Hoff and Dr. Betz, by use of the Froude momentum theory, had arrived at a value of 0.59 for maximum windmill efficiency as expressed in the equation:

$$\text{Efficiency} = \frac{\text{Power of Windmill}}{\text{Kinetic Energy of Air Cylinder of Equal Diameter}}$$

The evidence was that this figure was too low, and arguments against its acceptance were the facts that the air affected by a windmill had a larger diameter than the windmill itself, and also that the resistance of a windmill might be much greater than $\rho V^2/2g$.

In speaking of gust energy, Mr. Fales pointed out that the conventional type of wind-recording instrument, such as the Robinson cup anemometer, gave an approximately mean value of the wind velocity but not a value which was accurate to use as a measure of the kinetic energy of the wind. Even if it had no weight and no moment of inertia so that it recorded the true instantaneous velocity of the wind, its integrated readings would not give a true measure of the energy obtainable from the wind since the latter was dependent upon the sum of the cubes of the instantaneous velocities. As a matter of fact, the Robinson cup anemometer was not weightless, and in general Mr. Fales thought that Weather Bureau wind velocities could not be directly used as a means of determining the available energy of the wind, or the intensity or periodicity of the gustiness of the wind. He quoted from Langley's The Internal Work of the Wind as follows:

"The wind, starting with the velocity of 23 m.p.h., rose within 10 sec. to a velocity of 33 m.p.h., and within 10 sec. more fell to its initial speed. It then rose within 30 sec. to a velocity of 36 m.p.h., and so on, with alternate risings and fallings, at one time actually stopping; and, as the reader might easily observe, passing through 18 notable maxima and many notable minima, the average interval from a maximum to a minimum being a little over 10 sec., and the average change of velocity in this time being about 10 m.p.h."

Mr. Fales pointed out that the quantity of energy in a wind such as described by Langley would be considerably greater than that which would be indicated if the Robinson cup anemometer value of 23 m.p.h. were used. Langley's special instrument showed the rapid wind alterations, which the Robinson cup anemometer would not, and the power of a wind blowing for one unit of time at 33 m.p.h. and the next unit of time at 13 m.p.h. was 50 per cent greater than a 23-m.p.h. wind blowing for two units of time. Ordinary anemometer records would therefore lead us to underestimate the available power to be obtained from the wind.

[1] Aeronautical Engineer, Air Service, War Department, Dayton, Ohio.

Further consideration of wind-velocity fluctuations led to interesting conclusions regarding the design of windmills. Mr. Fales demonstrated by rough calculations that if a windmill were to avail itself to the full of the gust energy, it must follow each change in $V$ with a corresponding change in r.p.m. Fig. 10 was a performance chart of a windmill tested in the wind tunnel, showing the power characteristics plotted against tip-speed ratio. It was seen that for best operation the power coefficient was $1.0 \times 10^{-6}$, and the tip-speed factor 1.3. If, due to a gust, the wind suddenly changed from 23 to 33 m.p.h., and if the wheel speeded up immediately to keep step, it was obvious that the power absorbed would increase as the cube of the velocity, i.e., in the ratio of $(33)^3$ to $(23)^3$. However, if the wheel was so heavy and had so great an inertia that it did not speed up at all, the situation was as follows: The tip-speed factor did not remain at 1.3, but changed to 0.906, and the corresponding power coefficient was therefore only $^9/_{10}$ as great as before. The conclusion from this brief analysis was that a light windmill had an aerodynamic advantage over a heavy windmill entirely aside from cost of production. The analysis applied to the case of the common reciprocating type of windmill wherein the ratio of wind velocity to tip velocity was by no means constant or of the best value, for which reason a reciprocating-pump load was not favorable to maximum efficiency in a windmill.

In referring to a new basis of windmill design, Mr. Fales pointed

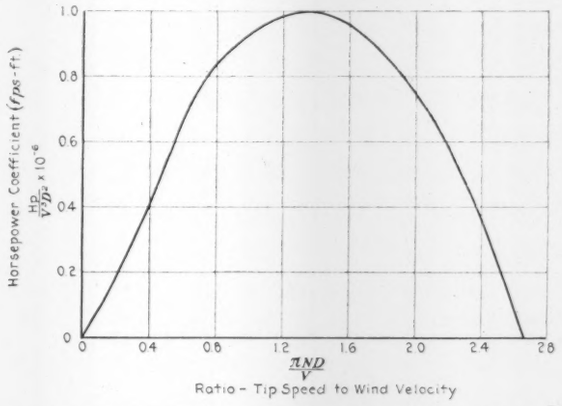


FIG. 10 HORSEPOWER COEFFICIENTS PLOTTED AGAINST RATIO OF TIP SPEED TO WIND VELOCITY AS OBTAINED IN WIND-TUNNEL TESTS OF WINDMILL

out that as a result of windmill applications during the war in driving gasoline pumps, cameras, generators, etc., on aircraft, a new theory had been developed on an aerodynamic basis which gave fair satisfaction for certain ranges, although blade interference did not allow aerodynamic coefficients to be used with complete accuracy. A large number of tests in the wind tunnel had, however, given valuable semi-empirical data.

Walter C. Durfee[1] criticized the paper as representing very special calculations, applicable to a particular pumping plant, a particular type of windmill, and a particular type of weather which must be mutually adjusted. He thought that the paper had value as showing the large number of varied problems encountered in this work. He did not see why the many-bladed or large-area type of windmill had been selected, and thought that the same power might be obtained at double the speed with half the torque and one-quarter the number of blades. Mr. Durfee pointed out that the windmill might be made to run safely in stormy weather if sufficiently stiffened, and that self-adjusting blades might also be designed usefully. He was in great sympathy with the work of the author from the view point of practical utility, apart from the theoretical discussion.

O. P. Hood[2] was glad to see the paper referring to a study of the available energy as a seasonal matter. In pumping water, for instance, with high wind velocities plenty of power might be available; yet in summer, when water was most needed, the wind velocity might be too low to be effective.

N. D. Stevens,[3] who was associated with Mr. Pancratz in his tests, emphasized the importance of using windmills which could work on very low wind velocities, available nearly all the time. Windmills utilizing high wind velocities would involve such ponderous apparatus that they would be useless for the more common low wind velocities.

H. M. Lane[1] described early experiments using air compressors driven by windmills, and spoke of the possibility of utilizing wind power in the mesas of the West, where whirlwinds or "dust devils" might raise special problems in getting sufficient strength for the windmills.

E. P. Warner[2] said that in undertaking the application of propeller theory either to the design of propellers themselves or to windmills, blowers, fans, or other machinery of that nature, three principal methods of attack were available. The analysis might be based solely on aerodynamic data obtained in the wind tunnel, the blade element method devised by Drzewiecki being used; it might rest purely on dynamic considerations derived from the work of Rankine and Froude and independent of experimental constants; or it might involve a combination of the two along the lines worked out by Fage, Riach, and others. The third method was the one now used in virtually all propeller design. The author appeared to have depended rather upon the second, for, although the availability of wind-tunnel figures was mentioned, there was little evidence that they had been put to use wherever they might have proved of service, and most of the reasoning seemed to be quite general in its nature and wholly independent of wind-tunnel results.

A concrete instance of this was found in the determination of the rate at which the air flowed into the wheel. The probable rate of flow across the several elements of a propeller blade could only be found by determining the thrust obtained from each element in turn by each of the two methods outlined above and equating the two results, thus obtaining an equation with the inflow as the sole independent unknown. One side of the equation contained constants obtained in wind-tunnel tests of the particular blade sections employed, and the actual form of the curve in Fig. 4 would therefore depend on the contour of the blade, its width, and the sectional form. The average velocity of inflow could be found directly if the total resistance of the wheel was known or assumed, and in actual practice it had been found that little if anything was gained in propeller-design practice by trying to calculate a new value of the inflow factor for each station along the blade instead of using an average figure and assuming it to remain constant.

In dealing with the subject of interferences between the blades the author appeared to have been betrayed into an incorrect or at least incomplete conclusion by his avoidance of wind-tunnel data. The subject of interference between aerodynamic surfaces placed in close tandem combination, as were the successively passing blades of a windmill wheel, was one which no theory completely covered, and which must ultimately be referred to the wind tunnel for investigation, although the recent work of Prandtl and Munk and Glauert had shed considerable light upon the phenomena. It required no experiment, however, to say that the statement that "interferences increase when the wind velocity increases above a certain value" was definitely misleading. If the wind velocity and the rate of rotation rose at the same rate, as they would tend to do if the windmill were driving a pump, for example, and was ungoverned, the interference would always represent a constant fraction of the total power developed, and the available torque would vary directly as the square of the wind speed. Absolute speeds had no significance in fixing the coefficients of performance of a given propeller or windmill. The slip function, or ratio of wind speed to the product of revolutions in unit time and diameter, was the only factor of importance in that connection. It was, to be sure, true in general that if the wheel were already governed so that the r.p.m. increased less rapidly than the wind speed, the ratio of interference to total power would in general tend to rise with rising wind velocity. Interference between the blades was not a factor of much importance in the old Dutch four-bladed type of windmill, but it was vital in modern American multi-bladed wheels.

The final problems of the propeller and the windmill were somewhat different. The airplane propeller was ordinarily chosen for

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high efficiency, and the diameter required to give a maximum of thrust from a stated torque was a factor of secondary importance. In a windmill, on the other hand, efficiency was neglected. There was always plenty of air, and the amount of energy abstracted from that which passed through the wheel did not matter in the least. The objects were to secure a high torque with a small diameter, and, most important of all in most cases, to get good regulation or governing.

The conditions of maximum torque varied with the speed at which the wheel was to run and the wind velocities expected. If such curves as those in Mr. Pancratz's Fig. 8 were plotted against $\theta + \alpha$ instead of against $\theta$ it would be found that when the r.p.m. were relatively low, and $\theta + \alpha$ accordingly large, $\alpha$ itself should be large for maximum torque, but that with a small value of $\theta + \alpha$ it was necessary to make $\alpha$ comparatively small in order that the wheel might run at all. The exact figures would depend on the section used.

Good regulation might imply a variety of different things, depending on the type of machinery to be driven. The simplest solution was found in the use of mechanisms capable of running at substantially any speed and having a torque varying rapidly with speed, such as pumps. The torque required to drive a pump being approximately proportional to the square of the r.p.m., the rate of rotation of the wheel would vary in direct proportion to the wind speed if no governing device were used. A second case arose when the windmill drove a generator with third-brush regulation, the rate of charging and the load on the wheel thus being kept nearly constant over a considerable range of speeds. The r.p.m. would then rise faster than the wind velocity, and might run up, in very high winds, to fifteen or twenty times the speed at which charging began. In order to get the maximum profit out of winds above the minimum, it might be advisable to have a second generator cutting in at a somewhat higher speed than the first.

In case the machinery to be driven was of a constant-speed type the problem of governing became very difficult. In order that there might be any stable rate of rotation it was necessary that the torque developed should increase as the r.p.m. decreased at a constant wind velocity, and that, in turn, required an angle of attack somewhat below that of maximum torque. It obviously followed that the torque at constant r.p.m. must change with wind speed at a rate somewhat higher than the square of the speed. If a wheel were to run at the same speed at all wind velocities from, say, 5 to 60 m.p.h., the braking torque must be some 500 times as large at the highest speed as at the lowest. No ordinary method of variable-load governing would suffice to cover such a range. It therefore became necessary to resort to the common expedient of pivoting the blades of the wheel and governing by varying their angles of attack.

Finally, a word might be said about the stresses in windmill wheels, although that subject was not dwelt upon at length in the author's paper. If a mill were to be erected in a country liable to violent storms it was necessary either to make the wheel very heavy to stand the stresses arising at high speeds or to provide some means of relieving those stresses. The matter was particularly critical in ungoverned wheels used for pumping, as an increase of wind velocity augmented both the tip speed and the air load normal to the vanes in those cases. Perhaps the best expedient was to reinforce the vanes to carry very high loads in pure tension and then to provide an automatic cut-off disconnecting the pump from the wheel, or a bypass or some equivalent mechanism insuring a rapid decrease of the load substantially to zero after a certain wind speed was reached. When the load was thus removed the wheel would develop only enough torque to overcome friction. The blades would work at an angle of attack very near to that of zero lift, and the normal pressure on the vanes would therefore fall to a very small value. The wheel would race up to an exceedingly high speed, but the only important stress would be the pure tension due to centrifugal force, and that was comparatively easy to carry if the tip angle of the blade was 12 deg., as Mr. Pancratz said it must be to give sufficient starting torque, as the tip speed could then never exceed five times the wind speed.

In closing Mr. Pancratz said that he thought Mr. Fales' discussion very interesting, especially the power curve resulting from a test on a multi-blade model windmill in a wind tunnel. Mr. Fales had mentioned the results obtained by Doctors Hoff and Betz, who arrived at a value of 59 per cent for the maximum windmill efficiency. The test on the 15–8-ft. test wheel showed a maximum efficiency of 64 per cent as expressed by the equation:

$$\text{Efficiency} = \frac{\text{Power to Generator}}{\text{Kinetic Energy of Air Cylinder } \pi/4\,(15^2\text{–}8^2)}$$

The maximum, 64 per cent, was derived from tests in a steady wind at 7.6, 7.8, 8.3 and 8.4 m.p.h. During these tests the charging rate to battery was constant. It was difficult to set a value for the losses due to the three sets of gears, friction of ball bearings, and the drag of the necessary vane braces of the wheel. It was fair to estimate that the efficiency of the vanes was over 70 per cent. Recent results of tests of a wind-generator plant indicated that even a higher maximum efficiency than 70 per cent was feasible for a commercial unit. The power curve of the multi-blade windmill indicated a maximum efficiency of 60 per cent for all wind velocities and sizes of wheel. The 15–8-ft. test wheel ran 23 r.p.m. under maximum load in a steady 6 m.p.h. wind. The value of $\frac{\pi ND}{V}$ in this case was 2.05. From the curve the coefficient was 0.72, giving an efficiency of 43 per cent as the maximum for this wheel. This was considerably lower than the 70 per cent of the 15–8-ft. wheel. Consequently the power curve of a small model derived from a wind-tunnel test was not a correct curve for the output of large wheels in low wind velocities. At present, it was difficult even to estimate what the maximum was for the 13-ft. wheel. But in the near future the results of a static wind-tunnel test might indicate what could be expected as to efficiency, power, and control. Corrections would have to be made in applying these data in the design of a full-sized wheel. The final test of a full-sized wheel over a long period of time would give the definite data and show its commercial value.

High efficiency was desirable, but the important factor to consider was the development of the maximum power per square foot of vane surface. This was necessary to get the best results, considering storm conditions, the varying velocities of the wind, and the cost of manufacturing.

Apparently, continued Mr. Pancratz, Mr. Fales had made a very careful study of gust energy. It was unfortunate that the windmill would never be able to utilize the gain in available energy due to the rapid rising and falling of velocities which occurred in nearly all winds, especially those above 9 m.p.h. The flywheel effect of the wheel and the driven revolving parts would always be too high.

It was very advantageous to keep the flywheel effect low in value, but there was a considerable gain in the power from the slow rising and falling of the wind. The U. S. Weather Bureau gave an average wind velocity of 9 m.p.h. for Chicago. If in a cycle of 11 hr. the wind gradually rose from 4 miles to 14 miles maximum and decreased gradually to 4 m.p.h., the average was 9 m.p.h. The cube root of the mean cube equaled 10 m.p.h. Available energy varied as the cube. This gave a gain of 37 per cent available over a steady 9-mile wind. This gain due to the gradual changes in wind strength was very noticeable in the test of the 15–8-ft. wheels, especially in winds above 9 and 10 m.p.h.

## Bearing Alloys with Low Copper Content

THE cost of copper in Germany has prompted the machine-tool makers to search for bearing alloys with a low copper content. Extensive experiments with six different alloys have been made in Berlin with both sand and chilled castings. It was found that neither the pressure in the bearing nor its running temperature was the decisive factor in determining the life of a brass. Contrary to the expectation of the shop experts, bushings cast on sand cores were distinctly better than those cast on iron cores. When broken, these bushings showed that the structure of the fractures was not very different for the two ways of casting. The conclusion reached was that bushings with a high zinc content ought to be cast on sand cores and lubricated with a good oil. They were found to expand more than do bronze bushings, and need a little extra play for the axle. It is to be noted that the actual proportion of copper to zinc has not been disclosed in the continental technical press. *The Engineer*, Sept. 19, 1924, p. 323.

# Limiting Cases in Involute Spur Gearing

By A. B. COX,[1] EAST PITTSBURGH, PA.

*In this paper the author derives formulas by means of which it is possible to determine the smallest number of teeth that an involute spur gear can have and yet be a perfect gear, i.e., have teeth formed by straight, non-undercut flanks and unmodified involute faces, and be capable of rotating in either direction, during which rotation at least one tooth of the driving gear will always be in contact with a tooth of the driven gear so that the rotation of the driven gear will be continous and uniform.*

*Three sets of limiting cases are discussed, namely, for 1-to-1 gear ratio, for 1-to-∞ (rack and pinion) ratio, and for a gear interchangeable in any ratio from one extreme (1 to 1) to the other (1 to ∞); and a new polar formula for the involute is presented which, it is believed, will serve as a valuable tool for the further analysis of gearing.*

IT IS an axiom that when we know the limitations of a piece of apparatus, we are—other things being equal—best qualified to judge as to what are possible or desirable standards for the apparatus in question. In the discussions of standardization of gearing there is so much said nowadays in favor of a larger angle for the line of action in involute gears and of a smaller depth of tooth that one can hardly be blamed for beginning to wonder whether there are limits to such changes, and if there are limits, how close the old standards are to them. Since a very little study of the subject indicates that there are limits, this paper has been prepared in an effort to determine just what and where these limits are.

The mathematical basis for this paper appeared in the *American Machinist* in the first three of a series of five articles on involute gearing. These articles dealt respectively with interference (Oct. 14, 1920), number of teeth in contact (Nov. 11, 1920), and angles of approach, recess, and contact (Mar. 31, 1921). For the convenience of the readers of the present paper a very short résumé of this previously published work will be given. Where the ± sign has been replaced by + or by − in the formulas, the proper sign has been determined by actual trial. The following notation is employed.

$Y$ = (See Fig. 1)
$n$ = number of teeth in pinion
$N$ = number of teeth in gear
$K$ = $N/n$ = gear ratio
$P_c$ = circular (or linear) pitch
$P$ = diametral pitch
$P_t$ = pitch of teeth along line of action
$A$ = length of addendum
$q$ = $A/P_c$
$a$ = radius of base circle
$r$ = radius of pinion pitch circle
$R$ = radius of gear pitch circle
$R_0$ = radius of gear addendum circle
$R_b$ = (See Fig. 3)
$\rho$ = radius vector of involute (with subscript for individual values)
$H$ = $\rho/r$
$L_p$, $L_r$, $L_t$ = portions of line of action (See Fig. 2)
$M_p$ = number of pairs of teeth in contact on pinion side of line of centers
$M_r$ = ditto on rack side of line of centers
$M_t$ = total number of pairs of teeth in contact in a pair of meshing gears
$\phi$ = angle of line of action (pressure angle)
$\alpha$ = angle of radius vector of involute (with corresponding subscript)
$\theta$ = (See Fig. 3)
$\beta$ = $\tan^{-1}\theta$
$\psi$ = angle subtended by $^1/_4$ circular pitch.

[1] Navy Dept. of U. S. Govt., at Westinghouse Elec. & Mfg. Co.

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## INTERFERENCE

It is well known that in order to avoid interference in a pair of meshing involute gears having non-undercut teeth, the teeth of either gear must make contact with the teeth of the other gear only on the curve of the involute—never on the straight flanks of the teeth. Since the involute forming the curved part of the tooth extends down only as far as the base circle, and since it is characteristic of involute gears that the teeth of meshing gears make contact only at points on the line of action, the limiting point at which the teeth may begin to make contact with each other without interference is at the point of intersection (more properly, the point of tangency) of the line of action with the base circle. And since the pinion always has a greater length of straight flank than the mating gear, interference will occur, if at all, first and most seriously where the ends of the gear teeth touch the straight flanks of the pinion teeth. The limiting case for non-interference is therefore obtained when the teeth are of such number and proportions that the addendum circle of the gear passes through the point of tangency of the line of action and the base circle as at point $b$ in Fig. 1.

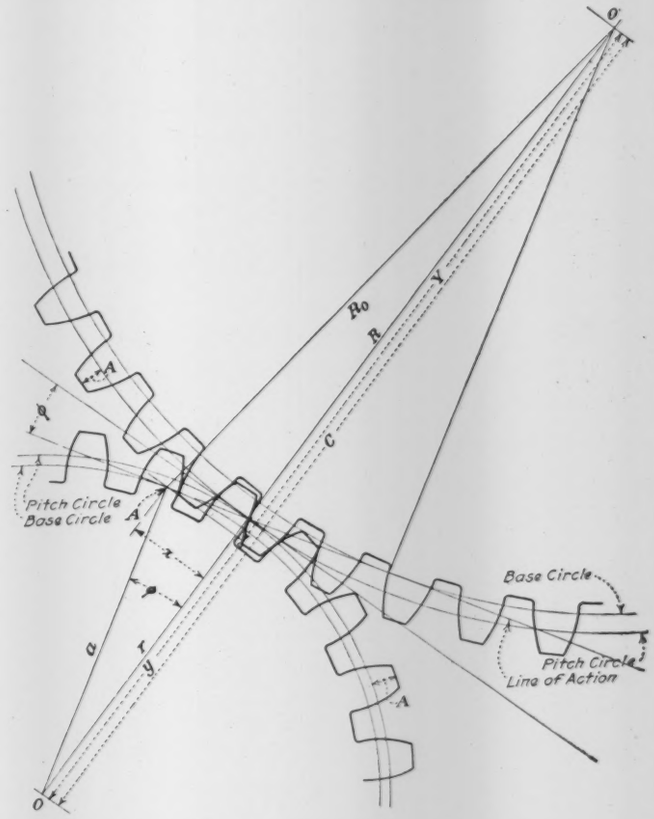


FIG. 1 INTERFERENCE DIAGRAM

Taking the origin of a system of rectangular coördinates at $O$, the center of the pinion, it is possible to obtain an equation involving the radius of the base circle of the pinion, the radius of the addendum circle of the gear, and the distance between centers of the gears, which when solved gives the smallest number of teeth possible to have in a pinion having teeth of a given depth and given pressure angle and meshing with a similar gear having the same or a larger number of teeth and yet avoid interference or undercutting. The derivation is as follows:

Referring to Fig. 1,

$$a = r \cos \phi \quad [1]$$

$$x = a \sin \phi = r \sin \phi \cos \phi \quad [2]$$

For convenience the length of addendum $A$ should be expressed as a fraction $q$ of the circular pitch $P_c$, or

$$A = qP_c \quad [3]$$

But since

$$P_c = \frac{2\pi r}{n} \quad [4]$$

[3] becomes

$$A = \frac{2\pi qr}{n} \quad [5]$$

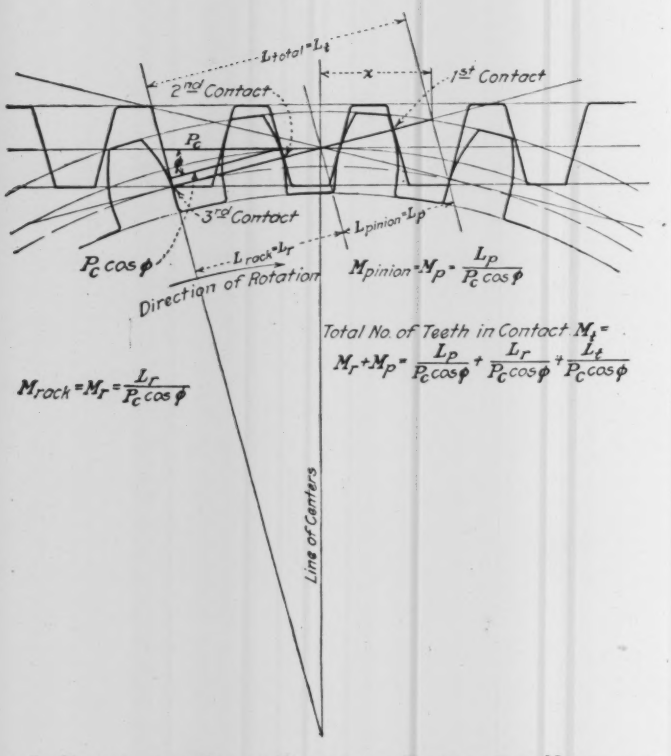


FIG. 2 DIAGRAM FOR USE IN DEVELOPING FORMULA FOR NUMBERS OF PAIRS OF TEETH IN CONTACT IN A PAIR OF MESHING GEARS
(Rack and 32-tooth pinion; 2-pitch standard 14½-deg. tooth.)

$$R_0 = R + A = \frac{N}{n}r + qP_c = Kr + \frac{2\pi qr}{n} \quad [6]$$

In the two right triangles formed by $a$, $x$, $y$ and $R_0$, $x$, $Y$, there are the relations

$$\sqrt{a^2 - x^2} = y \quad [7]$$

and

$$\sqrt{R_0^2 - x^2} = Y \quad [8]$$

But

$$Y + y = R + r = Kr + r = r(K + 1) \quad [9]$$

or

$$\sqrt{a^2 - x^2} + \sqrt{R_0^2 - x^2} = r(K + 1) \quad [10]$$

Substituting in [10] the value of $a$ from [1], of $x$ from [2], and of $R_0$ from [6],

$$\sqrt{r^2 \cos^2 \phi - r^2 \sin^2 \phi \cos^2 \phi} + \sqrt{[Kr + (2\pi qr/n)]^2 - r^2 \sin^2 \phi \cos^2 \phi} = r(K + 1) \quad [11]$$

Simplifying this expression and solving for $n$,

$$n = \frac{2\pi q[K + \sqrt{K^2 + (2K + 1)\sin^2 \phi}]}{(2K + 1)\sin^2 \phi} \quad [12]$$

The minimum number of teeth $n$ necessary in a pinion in order to avoid interference or undercutting is seen from Formula [12] to be dependent upon the gear ratio, the pressure angle, and the length of the teeth. For standard-length teeth $q = 1/\pi$, so that [12] becomes

$$n = \frac{2K + 2\sqrt{K^2 + (2K + 1)\sin^2 \phi}}{(2K + 1)\sin^2 \phi} \quad [13]$$

In ordinary external spur gearing the two extreme gear ratios are the 1-to-1 ratio and the 1-to-$\infty$ ratio (rack and pinion). For these two extreme ratios Formula [12] becomes

$$n = \frac{\pi q\,(1 + \sqrt{1 + 3\sin^2 \phi})}{1.5 \sin^2 \phi} \quad [14]$$

and

$$n = \frac{2\pi q}{\sin^2 \phi} \quad [15]$$

respectively. These two formulas are useful in the later work of determining the limiting cases which are the subject of this paper.

## NUMBER OF TEETH IN CONTACT

The method of developing a formula for the number of pairs of teeth in contact in a pair of meshing gears is easily understood from the diagrammatic layout of Fig. 2. Since it is characteristic of involute gears that the teeth of the pinion make contact with those of the gear only on the line of action, all that is necessary is to find the length of the line of action included between the two addendum circles of the two gears and divide this length by the distance between adjacent teeth as measured along the line of action. For the gears shown in Fig. 2 this procedure results in the number 2.3 (approx.). This number means that during the rotation of these gears there will always be two pairs of teeth in contact and carrying the load, and that during approximately 0.3 of the time a third pair of teeth will be in contact and sharing the load. Obviously by shortening the teeth just a little it would be possible to get rid of the 0.3 so that only the whole number 2.0 would remain. This number means that just as one pair of teeth would be leaving contact, another pair would be coming into contact, so that there would always be just two pairs of teeth in contact and carrying the load at all times during the rotation of the gears. Obviously also, by carrying this process of tooth shortening far enough, the number of pairs of teeth in contact can be reduced to 1.0. This would mean that at all times during the rotation of the gears only one pair of teeth would be in contact and carrying the load; as soon as one pair of teeth started to leave contact another pair would begin to make contact. If continuous, uniform rotation of the driven gear is to be retained, the shortening process can of course be carried no further. This, then, is another limiting condition the variables of which must be definitely determined. In the derivation of a formula for the number of pairs of teeth in contact, instead of at once finding the total length $L_t$ of the portion of the line of action included between the addendum circles of the two gears, it is easier to find the length a part at a time and to obtain the total length by the addition of these parts. For example, the length $L_p$ is easily found by taking the center of the pinion, $O$, as the origin of a system of rectangular coördinates, finding the $x$ of the point of intersection of the line of action with the addendum circle of the pinion and multiplying the result by sec $\phi$.

The formula for the line of action is

$$y = x \tan \phi + r \quad [16]$$

and that for the addendum circle of the pinion is $x^2 + y^2 = (r + A)^2$, or, substituting [5],

$$x^2 + y^2 = \left(r + \frac{2\pi qr}{n}\right)^2 \quad [17]$$

By eliminating $y$ from these two simultaneous equations and solving for $x$, the abscissa of the point of intersection of the line of action with the addendum circle of the pinion is found to be

$$x = r \cos \phi \left[-\sin \phi + \sqrt{\sin^2 \phi + \frac{4}{n^2}\left(\pi nq + \pi^2 q^2\right)}\right] \quad [18]$$

The pitch distance between adjacent teeth as measured along the line of action is $P_c \cos \phi$, or, substituting [4],

$$P_t = \frac{2\pi r \cos \phi}{n} \quad \ldots\ldots [19]$$

Therefore the number of pairs of teeth in contact on the pinion side of the line of centers is $\frac{x \sec \phi}{P_c \cos \phi}$, or

$$M_p = \frac{n}{2\pi \cos \phi}\left[-\sin \phi + \sqrt{\sin^2 \phi + \frac{4}{n_2}\left(\pi nq + \pi^2 q^2\right)}\right] \quad [20]$$

In the 1-to-1 gear ratio the total number of pairs of teeth in contact is of course $2M_p$, or

$$M_t = \frac{n}{\pi \cos \phi}\left[-\sin \phi + \sqrt{\sin^2 \phi + \frac{4}{n^2}\left(\pi nq + \pi^2 q^2\right)}\right] \quad [21]$$

The length of the remainder of the line of action is $L_r = A \csc \phi$, or, substituting [3],

$$L_r = qP_c \csc \phi \quad \ldots\ldots [22]$$

and the number of pairs of teeth in contact on the rack side of the line of centers is

$$M_r = \frac{L_r}{P_c \cos \phi} = \frac{qP_c \csc \phi}{P_c \cos \phi} = \frac{q}{\sin \phi \cos \phi} \quad \ldots\ldots [23]$$

For the other extreme gear ratio (1 to $\infty$), therefore, the total number of pairs of teeth in contact is given by

$$M_t = M_r + M_p = \frac{q}{\sin \phi \cos \phi} + \frac{n}{2\pi \cos \phi}\left[-\sin \phi + \sqrt{\sin^2 \phi + \frac{4}{n^2}\left(\pi nq + \pi^2 q^2\right)}\right] \quad [24]$$

Formulas [23] and [24] will be used in derivations in connection with limiting cases later.

### Simplified Formula for the Involute

The equations of the involute, as ordinarily given, are so complex that it is necessary to derive a "new" formula therefor before it can be used in the derivations to follow. This derivation is very simple. Referring to Fig. 3, the length of the line *bc* is by definition of the involute equal to the length of arc *ac*, or $ac = \theta R_b$, $\theta$ being expressed in radians. Since *obc* is a right triangle, $\theta^2 R_b^2 = \rho^2 - R_b^2$, and

$$\theta = \sqrt{\frac{\rho^2 - R_b^2}{R_b^2}} \quad \ldots\ldots [25]$$

Now if the value of $\rho$ can be found in terms of $R_b$, as it usually can be, the value of $\theta$ can be determined for that value of $\rho$. Let $\rho$ be expressed therefore as $HR_b$ where $H$ is any fractional, whole, or mixed number as required. $H$ is then the ratio of the length of $\rho$ to the length of $R_b$, or

$$H = \rho / R_b \quad \ldots\ldots [26] \qquad \text{and} \qquad \rho = HR_b \quad \ldots\ldots [27]$$

Substituting [27] in [25],

$$\theta = \sqrt{H^2 - 1} \quad \ldots\ldots [28]$$

In dealing with gears, $R_b$ is of course the radius of the base circle and

$$R_b = r \cos \phi \quad \ldots\ldots [29]$$

If in dealing with gear problems, $\rho$ is expressed in terms of the pitch radius $r$, as is usually most convenient, [28] becomes

$$\theta = \sqrt{H^2 \sec^2 \phi - 1} \quad \ldots\ldots [30]$$

It is also seen in Fig. 3 that

$$\tan \beta = \theta R_b / R_b = \theta \quad \ldots\ldots [31]$$

or

$$\beta = \tan^{-1} \theta \quad \ldots\ldots [32]$$

and $\alpha = \theta - \beta$, or substituting [32]

$$\alpha = \theta - \tan^{-1} \theta \quad \ldots\ldots [33]$$

For the mathematical involute, therefore,

$$\alpha = \sqrt{H^2 - 1} \quad - \tan^{-1} \sqrt{H^2 - 1} \quad \ldots\ldots [34]$$

and for the involute as applied to gears,

$$\alpha = \sqrt{H^2 \sec^2 \phi - 1} \quad - \tan^{-1} \sqrt{H^2 \sec^2 \phi - 1} \quad \ldots\ldots [35]$$

### Limiting Cases

In general there may be said to be three sets of limiting cases in perfect involute gears: (1) the limiting case for 1-to-1 gear ratio; (2) the limiting case for 1-to-$\infty$ (rack and pinion) ratio; and (3) the limiting case for a gear interchangeable in any ratio from one extreme (1 to 1) to the other (1 to $\infty$). Since the limiting cases for single gear ratios other than the 1-to-1 and the 1-to-$\infty$ will of course lie between the results obtained for these two extreme ratios, the formulas and illustrations in this discussion are confined to these extreme ratios and to the interchangeable gear.

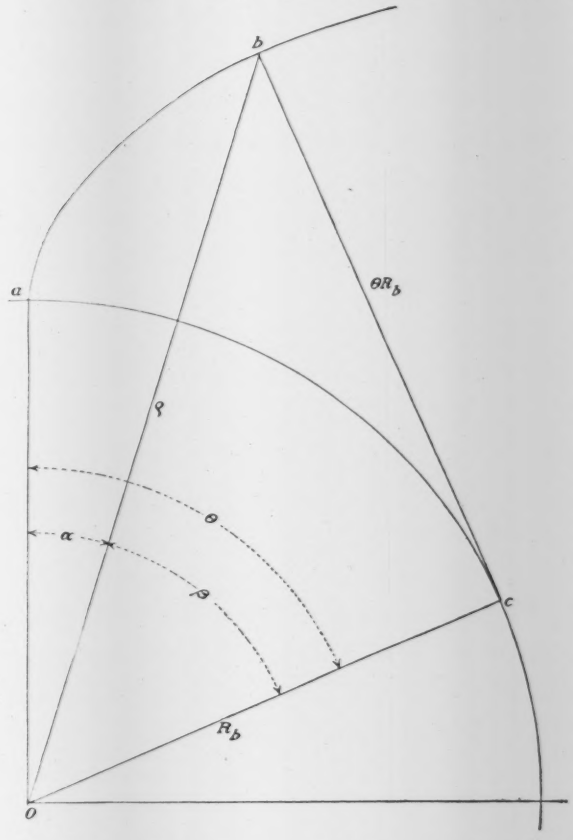


FIG. 3 DIAGRAM FOR USE IN DERIVING NEW FORMULA FOR THE INVOLUTE

The general limiting condition in each of these three limiting cases is best put in question form, viz., What is the smallest number of teeth that a gear can have and yet be a perfect gear, perfect in form as well as perfect in action? By "perfect in form" is meant a gear having teeth formed by straight non-undercut flanks and unmodified involute faces. By "perfect in action" is meant that the gear may rotate in either direction and that during the rotation, at least one tooth of the driving gear will always be in contact with a tooth of the driven gear, so that the rotation of the driven gear will be continuous and uniform. In order that the flanks of the teeth may be straight and that uniform rotation may be obtained, the conditions of *number of teeth, relative length of addendum,* and *angle of line of action* must not be such as to cause interference in gears of the ratio being considered, but must at the same time be such that one pair of teeth in contact will not leave contact before a second pair of teeth make contact. By properly combining the

interference formulas and the formulas for the number of pairs of teeth in contact in a pair of meshing gears, the formulas for the limiting cases of gearing can be worked out. The limiting cases of imperfect involute gearing will be dealt with separately after the perfect gears have been disposed of.

### 1-TO-1-RATIO LIMITING CASE

Solving [11] for $q$,

$$q = \frac{n}{2\pi}\left(-1 + \sqrt{1 + 3\sin^2\phi}\right) \quad [36]$$

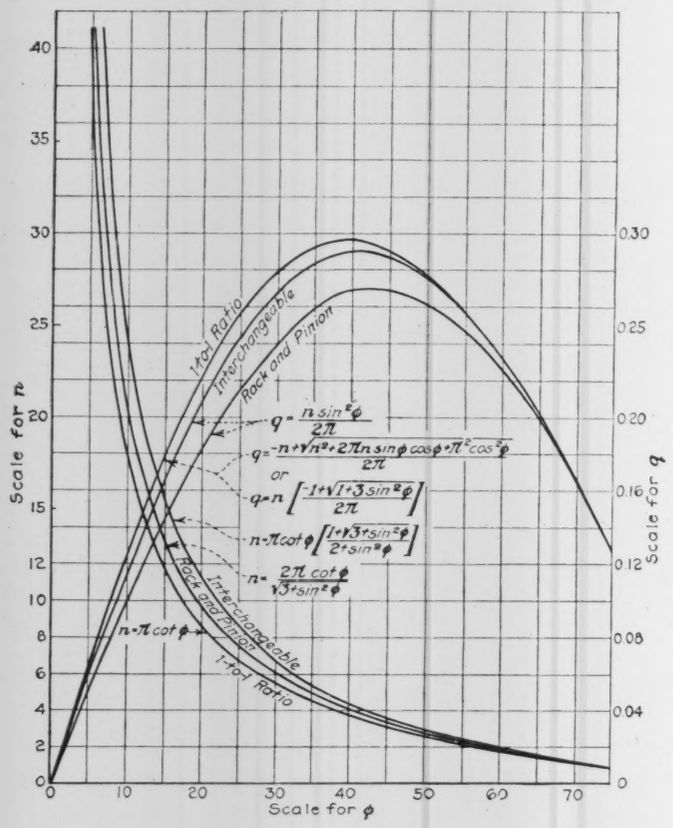
Scale for n
Scale for q
Scale for φ
1-to-1 Ratio
Interchangeable
Rack and Pinion
q = n sin²φ / 2π
q = -n + √(n² + 2πn sinφ cosφ + π² cos²φ) / 2π
or
q = n [-1 + √(1 + 3 sin²φ) / 2π]
n = π cotφ [1 + √(3 + sin²φ) / 2 + sin²φ]
n = 2π cotφ / √(3 + sin²φ)
n = π cotφ
Rack and Pinion
Interchangeable
1-to-1 Ratio

FIG. 4 LIMITING CONDITIONS FOR INVOLUTE GEARING

Solving [21] for $q$,

$$q = \frac{1}{2\pi}\left(-n + \sqrt{n^2 + 2\pi n M_t \sin\phi\cos\phi + \pi^2 M_t^2 \cos^2\phi}\right) \quad [37]$$

Equating these two values of $q$ and solving for $n$,

$$n = \pi M_t \cot\phi \quad [38]$$

and

$$\cot\phi = \frac{n}{\pi M_t} \quad [39]$$

If $M_t$ be taken as unity,

$$n = \pi \cot\phi \quad [40]$$

This is the formula which, for any number of teeth assumed for the gears, will give the proper angle of the line of action that will result in only one pair of teeth being continually in contact without interference as the gears rotate. The corresponding value of $q$ can be found from either [36] or [37]. Values of $n$ and $q$ have been plotted against values of $\phi$ (in degrees) in Fig. 4.

As smaller and smaller values of $n$ in Formula [40] are taken in an effort to find the least number of teeth possible to have in a perfect 1-to-1 ratio gear, the angle $\phi$ increases to a greater and greater value until finally it is found that for some certain value of $n$ the angle $\phi$ is so great that the teeth of the gears come to a point before reaching the value of $q$ required to give one pair of teeth continuously in contact. In Fig. 5, worked out for the interchangeable gear, it is easily seen that in the 1-to-1 ratio a given pair of teeth in contact will, as the gears rotate, leave contact with each other before the next pair following come into contact. There will then of course be a sudden pause in the rotation of the driven gear, so that the action of the gearing will not be perfect. It is clear, then, that the limiting condition occurs when the teeth are just long enough to enable one pair of teeth to hold contact until the following pair make contact without interference. It is easily seen from Formula [38] that the larger the angle of the line of action, the smaller $n$ becomes, so that $\phi$ must be taken as large as possible. An examination of the figures accompanying this paper will show that these limiting conditions occur when the teeth come to a point on the theoretical addendum circle as determined by Formula [40] and [36] or [37].

The actual work of derivation is better understood by reference to Fig. 6. In the triangle $obc$, $ob$ is perpendicular to $bc$, because it is drawn to the point of tangency of $bc$ to the base circle. Since $ob$ is a radius of the base circle, $ob = r\cos\phi$ [1]. And since $bc$ is the distance between adjacent teeth along the line of action, $bc = 2\pi r\cos\phi/n$, [19]. $\rho_1$ is therefore the square root of the sum of the squares of $ob$ and $bc$, or

$$\rho_1 = \sqrt{r^2\cos^2\phi + \frac{4\pi^2 r^2\cos^2\phi}{n^2}} \quad [41]$$

Simplifying,

$$\rho_1 = r\cos\phi\sqrt{1 + \frac{4\pi^2}{n^2}} \quad [42]$$

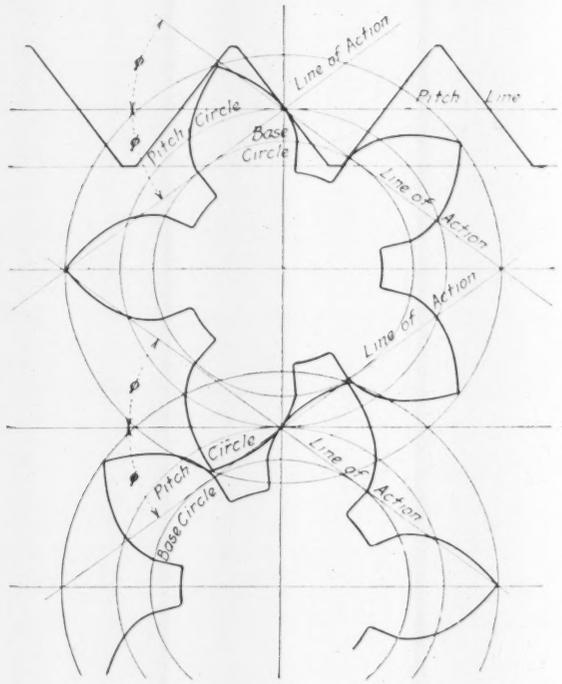


FIG. 5 SHOWING THAT IN 1-TO-1-RATIO GEARS A GIVEN PAIR OF TEETH LEAVE CONTACT BEFORE THE NEXT PAIR FOLLOWING COME INTO CONTACT ($n = 5$; $P = 1$; $d = 5$ in.; $\phi = 37$ deg.; $P_c = 3.1416$ in.)

By Formula [26]

$$H_1 = \cos\phi\sqrt{1 + \frac{4\pi^2}{n^2}} \quad [43]$$

Substituting [43] in [35]

$$\alpha_1 = \sqrt{\left(1 + \frac{4\pi^2}{n^2}\right) - 1} \quad - \tan^{-1}\sqrt{\left(1 + \frac{4\pi^2}{n^2}\right) - 1}$$

or, simplifying,

$$\alpha_1 = \frac{2\pi}{n} - \tan^{-1}\left(\frac{2\pi}{n}\right) \quad [44]$$

Similarly, $\rho_0$ for $\alpha_0$ is $r$, and therefore

$$H_0 = 1 \quad [45]$$

Substituting [45] in [35],

$$\alpha_0 = \sqrt{\sec^2 \phi - 1} - \tan^{-1} \sqrt{\sec^2 \phi - 1}$$

or, simplifying,

$$\alpha_0 = \tan \phi - \phi \quad [46]$$

Also

$$\psi = \alpha_1 - \alpha_0 \quad [47]$$

or, substituting [44] and [46],

$$\psi = \frac{2\pi}{n} - \tan^{-1}\left(\frac{2\pi}{n}\right) - \tan \phi + \phi \quad [48]$$

But since the arc of angle $\psi$ covers one-quarter of a tooth pitch, $\omega$ is equal to $2\pi/4n$, or

$$\psi = \frac{\pi}{2n} \quad [49]$$

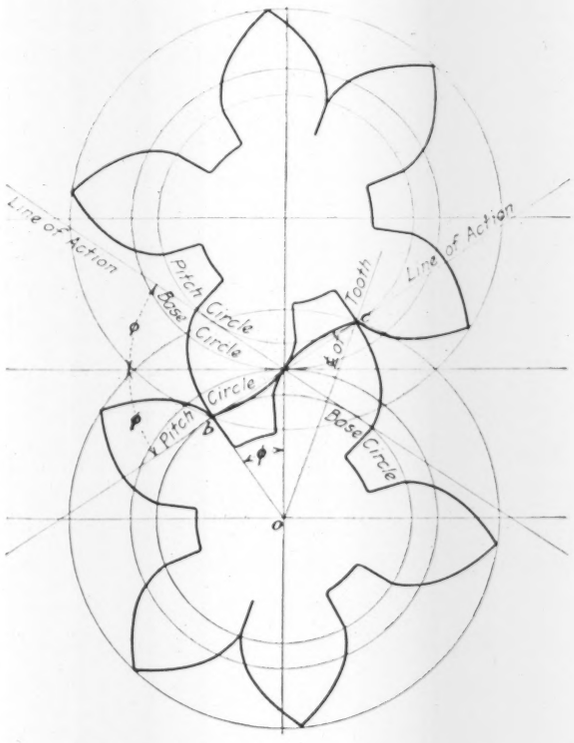


FIG. 6 DIAGRAM FOR USE IN DERIVING FORMULA
($n = 4.625$; $P = 1$; $d = 4.625$ in.; $q = 0.2920$; $\phi = 34$ deg. 12 min.; $P_c = 3.1416$ in.; $A = 0.917$ in.)

Equating [48] and [49],

$$\frac{2\pi}{n} - \tan^{-1}\left(\frac{2\pi}{n}\right) - \tan \phi + \phi = \frac{\pi}{2n} \quad [50]$$

Simplifying,

$$\frac{3\pi}{2n} - \tan^{-1}\left(\frac{2\pi}{n}\right) - \tan \phi + \phi = 0 \quad [51]$$

Substituting the value of $n$ from [40],

$$\frac{3\pi}{2\pi \cot \phi} - \tan^{-1}\left(\frac{2\pi}{\pi \cot \phi}\right) - \tan \phi + \phi = 0$$

Simplifying and rearranging terms,

$$\frac{\tan \phi}{2} + \phi - \tan^{-1}(2 \tan \phi) = 0 \quad [52]$$

This is an equation involving only one unknown ($\phi$), and may be solved by assuming values for $\phi$ and plotting the resulting value of the function on coördinate paper to determine the value of $\phi$ which will make the function equal to zero as required by [52]. The value of $\phi$ has been found by this method to be 34 deg. 12 min., and from Equation [40] the minimum value of $n$ was found to be 4.625, and the value of $q$ from [36] to be 0.292. Since it is obviously impossible to have a fractional number of teeth in a perfect gear, the next higher whole number, that is, 5, must be taken as the least practical

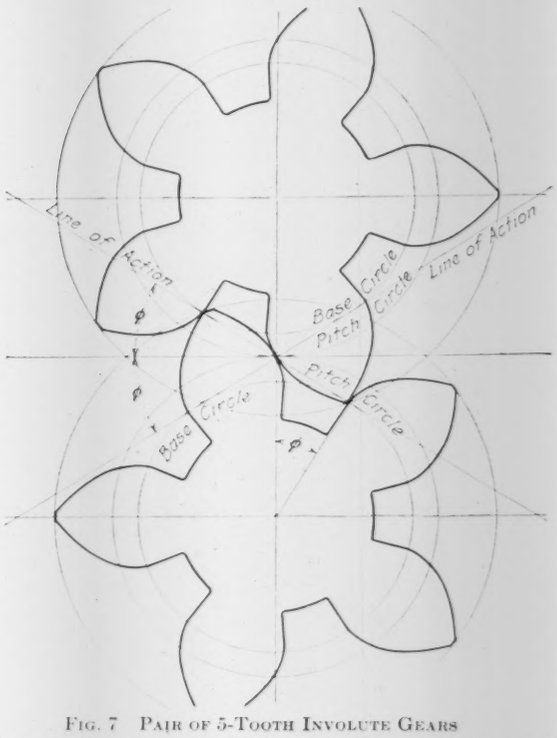


FIG. 7 PAIR OF 5-TOOTH INVOLUTE GEARS
($n = 5$; $P = 1$; $d = 5$ in.; $q = 0.2865$; $\phi = 32$ deg. 9 min.; $P_c = 3.1416$ in. $A = 0.900$ in.)

minimum number of teeth it is possible to have in a perfect involute gear driving perfectly in a 1-to-1 gear ratio. A pair of such 5-tooth gears is shown in Fig. 7. Fig. 6, showing gears with the fractional number of teeth, is given only for the purpose of demonstrating that the formulas actually do determine the theoretically limiting case. Figs. 8 and 10, the other two layouts which show gears with a fractional number of teeth, are given for a similar reason.

### 1-TO-∞ (RACK AND PINION) LIMITING CASE

The method of determining the limiting case for the 1-to-∞ ratio is similar to that just given for the 1-to-1 ratio and will therefore be given in a little less detail.

Substituting the value of $q$ from [15] in [24] and solving for $n$:

$$n = \frac{2\pi M_t}{\tan \phi \sqrt{3 + \sin^2 \phi}} \quad [53]$$

and when $M_t = 1$,

$$n = \frac{2\pi \cot \phi}{\sqrt{3 + \sin^2 \phi}} \quad [54]$$

The correspnding length of addendum is most easily found by solving [15] for $q$:

$$q = n \frac{\sin^2 \phi}{2\pi} \quad [55]$$

The values of $n$ and $q$ corresponding to various values of $\phi$ in [54] and [55] have been plotted in Fig. 4 along with similar quantities previously plotted for the 1-to-1 ratio limiting case.

As in the case of the 1-to-1 ratio, the limiting condition for the 1-to-∞ ratio occurs when the pinion teeth are pointed. In Fig. 8 the triangle $obc$ is a right triangle in which the same relations hold

as in triangle *obc* of Fig. 6. Formula [51] therefore holds for this limiting case also. Substituting the value of $n$ from [54], [51] becomes:

$$0.75 \tan\phi \sqrt{3 + \sin^2\phi} - \tan^{-1}(\tan\phi \sqrt{3 + \sin^2\phi}) - \tan\phi + \phi = 0 \quad [56]$$

This equation, as with [52], has to be solved by assuming values for $\phi$, expressing the second and fourth terms in radians and plotting the value of the entire equation obtained by substituting the assumed values of $\phi$ in [56]. The value of $\phi$ which makes this equation equal to zero has been found by this method to be 38 deg. 12 min., and from [54], $n$ was found to be 4.34, and the corresponding value of $q$ from [56] to be 0.264. As with the 1-to-1 ratio, instead of the fractional value of $n$ the next higher whole number must be chosen. This is 5 teeth, which is the same as the 1-to-1 ratio practical minimum. A perfect 5-tooth pinion is shown meshing with a perfect rack in Fig. 9.

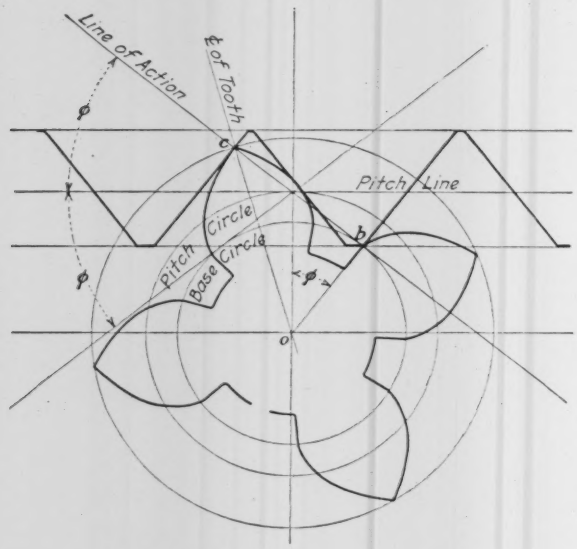


FIG. 8 RACK AND GEAR WITH FRACTIONAL NUMBER OF TEETH
($n = 4.34$; $P = 1$; $d = 4.34$ in.; $q = 0.2640$; $\phi = 38$ deg. 12 min.; $P_c = 3.1416$ in.; $A = 0.829$ in.)

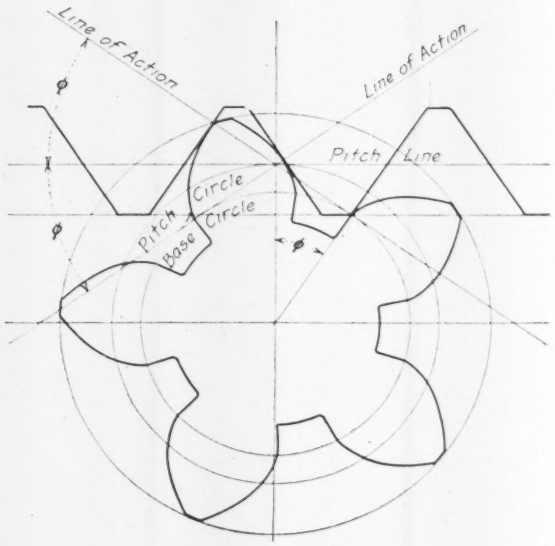


FIG. 9 PERFECT 5-TOOTH PINION MESHING WITH PERFECT RACK
($n = 5$; $P = 1$; $d = 5$ in.; $q = 0.2563$; $\phi = 34$ deg. 35 min.; $P_c = 3.1416$ in. $A = 0.805$ in.)

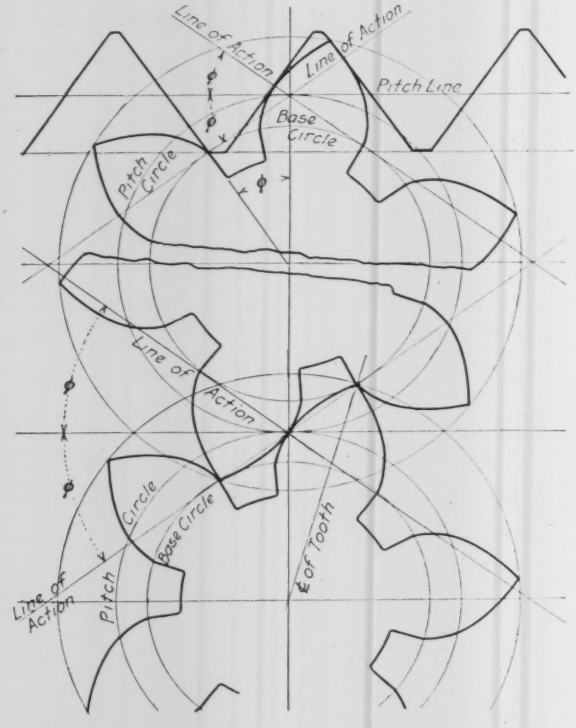


FIG. 10 DIAGRAMMATIC ARRANGEMENT OF TWO 1-TO-1 RATIO GEARS AND TWO 1-TO-∞ RATIO GEARS DESIGNED ACCORDING TO FORMULAS DERIVED
($n = 5.28$; $P = 1$; $d = 5.28$ in.; $q = 0.2860$; $\phi = 35$ deg. 42 min.; $P_c = 3.1416$ in.; $A = 0.898$ in.)

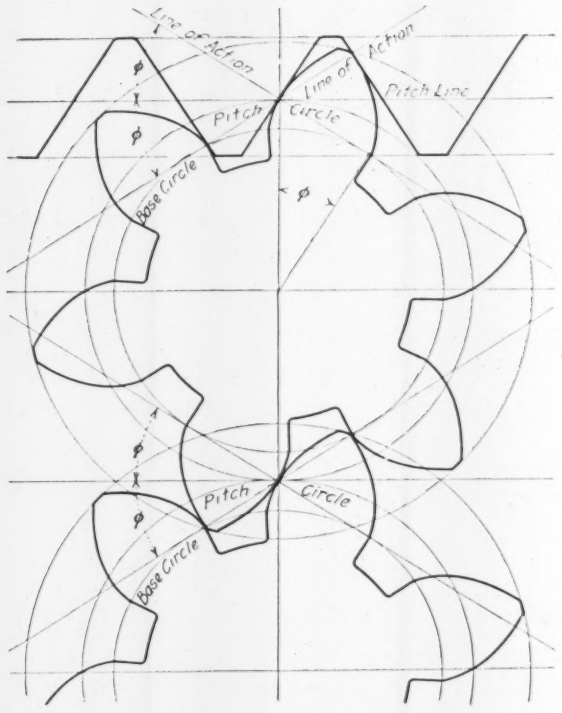


FIG. 11 SET OF PERFECT INTERCHANGEABLE GEARS WITH MINIMUM PRACTICAL NUMBER OF TEETH MESHING IN 1-TO-1 AND 1-TO-∞ RATIO
($n = 6$; $P = 1$; $d = 6$ in.; $q = 0.2794$; $\phi = 32$ deg. 45 min.; $P_c = 3.1416$ in.; $A = 0.878$ in.)

## INTERCHANGEABLE-GEAR LIMITING CASE

Since the theoretical minimum number of teeth for a perfect involute gear meshing in a 1-to-1 ratio is 4.625, and for a pinion meshing with a rack is 4.34, it would at first thought seem to be perfectly obvious that a 5-tooth gear of proper proportions ought to be interchangeable in either of these extreme gear ratios as well as in all ratios lying between. But a cut-and-try effort to design a perfect 5-tooth involute gear interchangeable in all gear ratios will soon convince one that it cannot be done.

The derivation of formulas for the accurate determination of the interchangeable-gear limiting case is slightly different from the derivations for the two limiting cases just given. From an examination of Fig. 10 it is easily seen that in the rack-and-pinion ratio the

limiting condition is not the number of pairs of teeth in contact but the interference condition, and that in the 1-to-1 ratio the limiting condition is not interference but is the condition that at least one pair of teeth shall be in mesh at all times during the rotation of the gears. Hence in order to find such proportions for the teeth that there shall be no interference in the rack-and-pinion ratio, while at the same time in the 1-to-1 ratio the tooth action shall be continuous, it is necessary to eliminate $q$ by equating Formula [55] from the interference formula for rack-and-pinion ratio to Formula [37] from the formula for number of pairs of teeth in contact in 1-to-1 ratio. Solving the resulting equation for $n$,

$$n = \pi \cot \phi \left( \frac{1 + \sqrt{3 + \sin^2 \phi}}{2 + \sin^2 \phi} \right) \quad \ldots\ldots [57]$$

Also in Fig. 10

$$\rho_1 = r + A \quad \ldots\ldots [58]$$

From [3] and [5] $A = qP_c = 2\pi qr/n$, or substituting the value of $q$ from [55],

$$A = r \sin^2 \phi \quad \ldots\ldots [59]$$

Substituting [59] in [58],

$$\rho_1 = r (1 + \sin^2 \phi) \quad \ldots\ldots [60]$$

Substituting [60] in [26],

$$H_1 = (1 + \sin^2 \phi) \quad \ldots\ldots [61]$$

Substituting [61] in [35],

$$\alpha_1 = \sqrt{(1 + \sin^2\phi)^2 \sec^2 \phi - 1} - \tan^{-1} \left( \sqrt{(1 + \sin^2\phi)^2 \sec^2 \phi - 1} \right) \quad \ldots\ldots [62]$$

Expanding [62] and substituting

$$(\sec^2 \phi - 1) = \tan^2 \phi \quad \ldots\ldots [63]$$

[62] becomes

$$\alpha_1 = (\tan \phi \sqrt{3 + \sin^2 \phi}) - \tan^{-1} (\tan \phi \sqrt{3 + \sin^2 \phi}) \quad [64]$$

From [46], $\alpha_0 = \tan \phi - \phi$; from [47], $\psi = \alpha_1 - \alpha_0$; and from [49], $\psi = \pi/2n$. Substituting [49], [46], and [62] in [47],

$$\frac{\pi}{2n} = (\tan \phi \sqrt{3 + \sin^2 \phi}) - \tan^{-1} (\tan \phi \sqrt{3 + \sin^2 \phi}) - \tan \phi + \phi \quad \ldots\ldots [65]$$

Substituting the value of $n$ from [57],

$$\left[ \frac{\tan \phi (2 + \sin^2 \phi)}{1 + \sqrt{3 + \sin^2 \phi}} \right] = \tan \phi \sqrt{3 + \sin^2 \phi} - \tan^{-1} (\tan \phi \sqrt{3 + \sin^2 \phi}) - \tan \phi + \phi \quad [66]$$

Partly clearing fractions and combining terms,

$$\left[ \frac{\tan \phi (2 + \sin^2 \phi)}{2 (1 + \sqrt{3 + \sin^2 \phi})} \right] + \phi - \tan^{-1} (\tan \phi \sqrt{3 + \sin^2 \phi}) = 0 \quad [67]$$

Solving [67] by the same methods that were used in the solution of [52] and [56], $\phi$ was found to be equal to 35 deg. 42 min. The value of $n$ was found from [57] to be 5.28, and the corresponding value of $q$ from [55] to be 0.286. Fig. 10 shows a diagrammatic arrangement of two 1-to-1 ratio gears and two 1-to-$\infty$ ratio gears designed according to these values of $\phi$, $n$ and $q$. It is easily seen from this figure that if $\phi$ were any greater the teeth would be too short to give continuous contact in the 1-to-1 ratio, and that if it were any smaller there would be interference in the 1-to-$\infty$ ratio. Fig. 10 therefore demonstrates that Formulas [67], [57], and [55] are correct, and that 5.28 is really the theoretical minimum number of teeth possible to have in a perfect interchangeable gear. Fig. 11 shows a set of perfect interchangeable gears with the minimum practical number of teeth, viz., 6, meshing in 1-to-1 and 1-to-$\infty$ ratio.

## LIMITING CASES OF IMPERFECT INVOLUTE GEARS

Any number of teeth from 6 on up can be used in perfect interchangeable involute spur gearing provided the limits for the tooth proportions set by the preceding formulas are not passed. Any attempt to go beyond these limits results in an imperfect or in a special gear. It has been shown that perfect 5-tooth gears can be designed but that they are not interchangeable. The 4- and the 3-tooth gears are the only other gears which can be designed so as to permit reversal of rotation and yet give continuous action. Of course these gears have to be undercut in order to get rid of interference and hence are not perfect gears as defined earlier in this

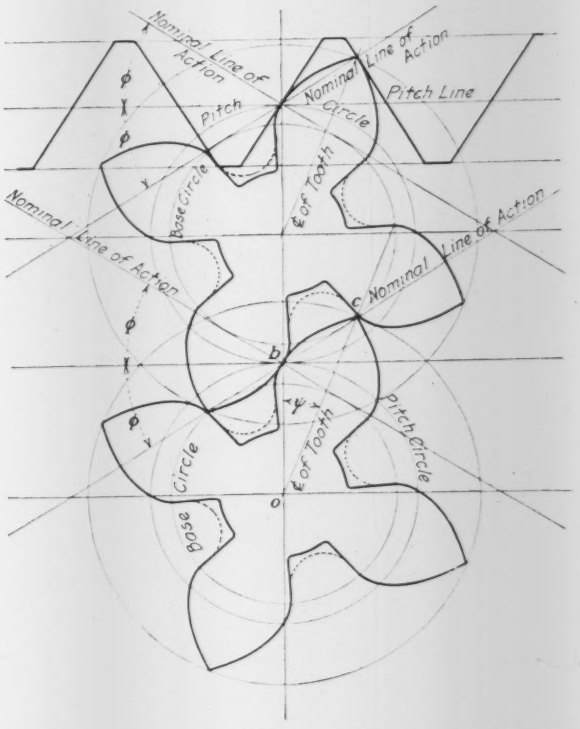


FIG. 12 MINIMUM AMOUNT OF UNDERCUTTING NECESSARY IN 4-TOOTH GEAR

($n = 4$; $P = 1$; $d = 4$ in.; $q = 0.2983$; $\phi = 32$ deg.; $P_c = 3.1416$ in.; $A = 0.937$ in.)

paper—that is, in these two gears all action between the teeth of meshing gears is not between the involute curves of the teeth.

There are no satisfactory formulas for determining beforehand the exact values of the variables $\phi$ and $q$ for either the 4- or the 3-tooth gear. Cut-and-try and approximate methods that are probably not of sufficient importance to be given in detail here were used in designing these gears.

*4-Tooth Gear.* An examination of Fig. 12 shows that for the 4-tooth gear a solution of the triangle *obc* would give the approximately correct value of $\phi$. In this triangle the side $ob = r$, and from [20], $bc = \pi r \cos \phi/n$, and the angle opposite one of them is known ($\psi = \pi/2n$). These do permit of the solution of triangle *obc* and the approximate determination of $\phi$. The minimum amount of undercutting necessary to avoid interference in the 1-to-1 ratio is shown in dotted line, and the minimum of undercutting for the condition of rack and pinion is shown in dot-and-dash line in the upper part of the figure only. The pressure angle at beginning contact between the teeth is about $35^1/_2$ deg. for the 1-to-1 ratio, and about $38^1/_2$ deg. for the rack and pinion. Of course the interference in the case of the rack and pinion can be got rid of by modifying the shape of the ends of the rack teeth, or by cutting off the ends of the rack teeth the required amount to accomplish the purpose. In the case rack teeth are modified by either of these two methods the undercutting of the teeth need not be greater than that necessary to avoid interference in the 1-to-1 ratio. The 4-tooth gear is therefore within the limits of possible workable design, but is a special gear.

*3-Tooth Gear.* The value of $\phi$ for the 3-tooth gear was determined by approximate and cut-and-try methods to be about $18^1/_2$ deg., and the corresponding value of $q$ to be about 0.350. From the layout

shown in Fig. 13 the amount of undercutting necessary for the 1-to-1 ratio is seen to be excessive and the undercutting for the rack and pinion still more so, although by cutting off or by shaping the ends of the rack teeth the amount of undercutting for both ratios can be kept down to that necessary for the 1-to-1 ratio. The pressure angle at beginning contact of the teeth is such as to make it more or less questionable as to whether such gears could be made to operate

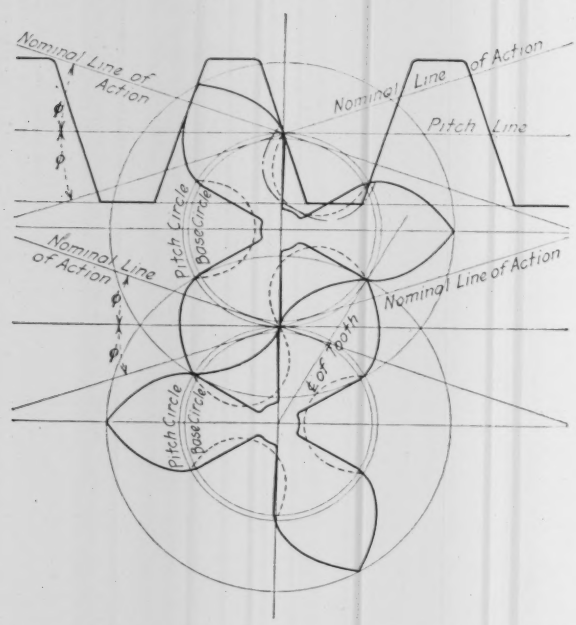


FIG. 13 SHOWING EXCESSIVE UNDERCUTTING IN THE CASE OF A 3-TOOTH GEAR
($n = 3$; $P = 1$; $d = 3$ in.; $q = 0.3497$; $\phi = 18$ deg. 30 min.; $P_c = 3.1416$ in.; $A = 1.098$ in.)

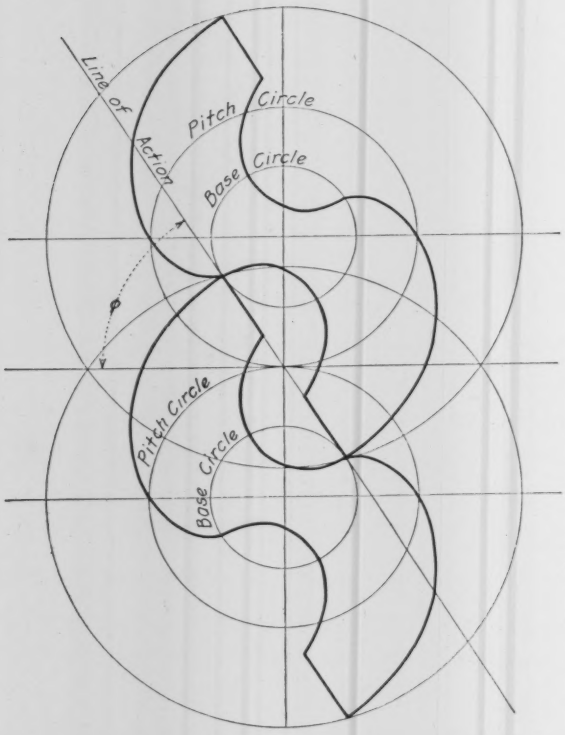


FIG. 14 LAYOUT FOR 1-TO-1-RATIO GEARS EACH WITH TWO TEETH
($n = 2$; $P = 0.5$; $d = 4$ in.; $q = 0.2457$; $\phi = 57$ deg. 31 min.; $P_c = 6.2832$ in.; $A = 1.544$ in.)

even without carrying any load. In any case the layout cannot be classed as a workable design.

*2- and 1-Tooth Gears.* For the 2- and 1-tooth gears no design of reversible gear, either with or without interference, is possible, but by using Formulas [39] and [11] it is easily possible to design perfect acting irreversible gears having these numbers of teeth. The layouts for 1-to-1 ratio gears having 2 teeth and 1 tooth, respectively, are shown in Figs. 14 and 15. The backs of the teeth in each case are formed by the minimum undercutting curve necessary to avoid interference at the back of the tooth.

## LIMITING CASES IN INVOLUTE SPUR GEARS CUT ACCORDING TO PRESENT STANDARDS

In the $14^1/_2$-deg. standard gear, $q = 1/\pi = 0.3183$, and with this length of addendum the minimum number of teeth possible to have in 1-to-1 ratio gears without undercutting to avoid interference is found by Formula [14] to be 23, and for the condition of rack and pinion is found by Formula [15] to be 32. Thirty-two is of course the minimum number of teeth for a gear interchangeable in any ratio from 1 to 1 to 1 to $\infty$ without interference or undercutting. With the shortest possible addendum which will just give continuous tooth contact, the minimum number of teeth a perfect-acting perfect

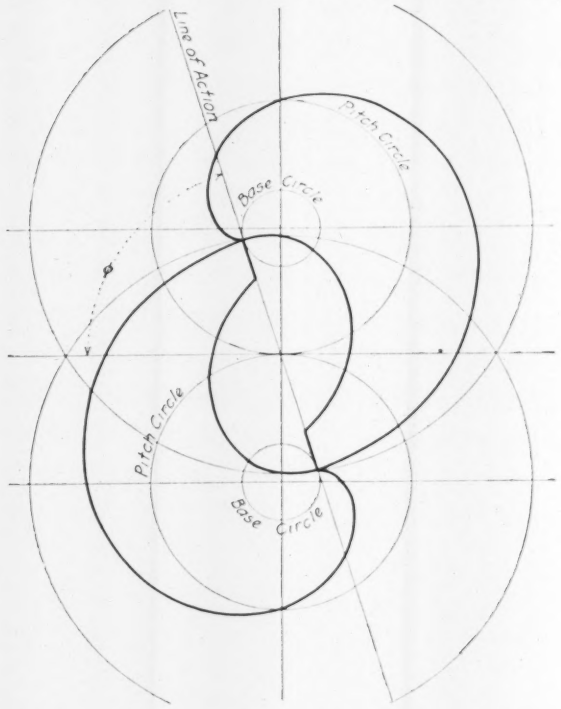


FIG. 15 LAYOUT FOR 1-TO-1-RATIO GEARS EACH WITH ONE TOOTH
($n = 1$; $P = 0.25$; $d = 4$ in.; $q = 0.1480$; $\phi = 72$ deg. 20 min.; $P_c = 12.5664$ in.; $A = 1.86$ in.)

$14^1/_2$-deg. involute gear interchangeable in any ratio from 1 to 1 to 1 to $\infty$ can have is found by Formula [57] to be theoretically 16.18, or practically 17.

In 20-deg. stub-tooth systems $q$ is usually made about 0.250, and the minimum values of $n$ without interference or undercutting are 9 and 14 for the 1-to-1 and rack-and-pinion ratios, respectively; 14 is also the minimum number of teeth for a gear interchangeable in all ratios. With the shortest practicable addendum, the minimum number of teeth a perfect-acting perfect 20-deg. involute gear interchangeable in any ratio can have is 12.

## DISCUSSION AND CONCLUSION

The curves of Fig. 4 are of particular interest because they show at a glance the possibilities of any particular existing or contemplated pressure angle. A comparison of the values obtained for the $14^1/_2$-deg. standard gear and for the 20-deg. stub-tooth gear with those of Fig. 4 is instructive because it can then be easily seen just how much farther it is possible to go in the changing of gear standards. Perhaps it is desirable to have two or three permanent standards to cover the entire practicable commercial range.

Particular care is necessary in the design of rolling-mill gears and other types of industrial gears which are subject to severe usage, and recalling the axiom in the introduction to this paper, we are best able to apply methods of design when we know their extreme possibilities. Before concluding, therefore, it is probably worth while

to hint at a few of the more useful design applications possible under the various divisions of the paper.

*Interference.* While the method of hobbing gears to avoid interference automatically is convenient, still it should be kept in mind that the undercutting weakens the teeth more than is necessary for small gear ratios. There is a possibility of increasing the strength of the teeth by using a larger fillet radius at the root. One concern has gone so far as to make the fillet a half-circle joining the flanks of adjacent teeth, and has claimed an increase in strength even though the depth of the space between the teeth was considerably increased.

*Number of Teeth in Contact.* In addition to being able to calculate the minimum depth of tooth necessary for continuous tooth action, it is very much more worth while to be able to determine the limiting conditions necessary in perfectly cut gears to avoid vibration with its consequent noise, inefficiency, increased wear, and breakage of the teeth. That vibration is recognized as a serious problem is evidenced by the following quotation from page 19 of a paper on The Modern Industrial Gear by W. H. Phillips and L. F. Burnham in the Proceedings of the Engineers' Society of Western Pennsylvania for February, 1923:

> The seriousness of vibration, whether of great or small amplitude, has always been recognized; and to no small degree vibration may be traced to the gear. The bearings that wear out too quickly, the shafts and pedestals that get out of alignment, the machine parts that become fatigued and fail, the motors that wear out commutators and brushes too quickly, and the armature leads that break—all these may be traced, in part at least, to vibration. Any formula figuring depreciation of equipment must be multiplied by a constant representing vibration.

There are two causes of vibration in perfectly cut gearing. First, there is the vibration of the individual teeth due to the recurrent loading and unloading of the teeth as they enter and leave the arc of contact during the rotation of the gears. This vibration involves only small masses, viz., only the mass of the gear teeth themselves, and is therefore relatively unimportant. Secondly, there is the vibration of the entire driven gear together with its entire load, this vibration being caused by the periodically varying number of load-carrying teeth. This vibration almost always involves large masses of material (the mass of the gear itself as well as the mass of the load being driven by the gear), and is therefore proportionately serious. When a gear is driving a varying load it is almost certain that at some particular value of the load the frequency of the variation of the number of load-carrying teeth will coincide with the natural period of vibration of the driven gear and its load. The noise of the gears for this resonant value of load is very bad, but practically disappears at loads above or below this critical value.

*New Formula for Involute.* The new form of the involute formula furnishes a tool for the further analysis of gearing. For examples of the use of this polar formula in gear analysis, see articles in vol. 54 (1921) of the *American Machinist* entitled Angles of Approach, Recess, and Contact (p. 559), and Slip of Involute Gear Teeth (pp. 913 and 951).

*Limiting Cases.* Aside from the establishing of limits for the various variables of gear design, it is expected that the chief value of the limiting cases discussed in this paper will be the stimulation of discussion and investigation of the principles of gearing by others.

## Discussion

LUTHER D. BURLINGAME[1] presented a written discussion in which he said he thought that the paper removed much of the uncertainty connected with the determination of limits of gear-tooth action. While he did not readily follow the derivation of some of the author's formulas, the graphic demonstration would seem to check them as being practically correct and therefore to be depended upon. He also thought that the deductions under the head of Limiting Cases in Involute Spur Gears According to Present Standards, while apparently correct, might be misleading, if they should leave the impression that present systems of 14½-deg. standard gears, interchangeable from a rack to a 12-tooth pinion, do not constitute a scientifically developed system giving one tooth or more in contact at all times and producing uniform motion. This whole matter had been gone into in much detail by the Committee on Standards for Involute Gears, appointed by the A.S.M.E. in 1908, and which had made its final report in 1913. It had been repeatedly brought up in the discussion that in order to retain the advantages of the 14½-deg. pressure angle, a portion of the tooth was modified from the true involute, and while this modification was along lines to give theoretically correct action, there was in addition an easing off at the point of the tooth for the purpose of producing quiet-running gears and such that manufacturing variations were provided for.

Mr. Burlingame believed that in addition to such gears, the successful use of which in interchangeable sets had been demonstrated through more than half a century, there was a field for gears with greater pressure angle and was interested to note that the author's figures checked with what had been the general understanding in the past, namely, that perfect-action 20-deg. involute gears could be made in interchangeable sets with a minimum of 12 teeth. Even here, however, the practical manufacturer had the problem of developing such modifications as would make it possible to ease off the point of the tooth for quiet running.

W. E. Sykes[1] wrote that the paper showed that the author had made a careful investigation of some of the basic features of involute gears. The expectation expressed in the last paragraph was a worthy one and the paper would in all probability stimulate the investigation of the principles of gearing by others. He did not wish to depreciate in any way the author's work, but he thought it was desirable to point out that a large amount of his labor was a repetition of what had been done before. Nearly all of the mathematical investigation set forth in the paper had been published by other workers. Similar formulas had been published in the English journal *Machinery*, and a series of articles by Mr. Arthur Fisher appeared, he thought, in 1919. In these articles the limiting cases were dealt with.

Several authors had published the new formula for the involute and this method had been applied to practical work for several years. In Mr. Tipple's book on gearing, published by the British Society of Shipbuilding and Engineering Draftsmen, a similar formula was given.

The author's advocacy of limiting the length of line contact was probably new, and was in Mr. Sykes' opinion only of academical interest. His argument as to its utility could only be sustained by assuming that gears could be accurately made. If gears were designed so as to have only one pair of teeth in contact, the slightest inaccuracy in manufacture would upset the whole principle of the design. If they were designed to have two pairs of teeth in contact, then any inaccuracy would result in two or more pairs being in contact at certain periods of revolution and only one pair of teeth in contact at other periods of revolution, and therefore no better results would be obtained in practice than those gotten by employing comparative haphazard methods of design.

Nearly all the limiting cases to which the author referred could be avoided by the use of helical or double helical teeth. For example, it was possible with double helical teeth to make a one-tooth pinion which would gear with wheels having any number of teeth and would transmit uniform motion in either direction. Such a one-tooth pinion had already been made and had been described in various publications. Practical applications of three-tooth pinions had been made and were in constant use. Moreover, the length of line contact could be easily fixed in herringbone gears so as to be at all times uniform, provided the gears were made with a suitable helix angle and a suitable width of face.

Mr. Sykes, however, did not agree that any definite length of line contact possessed peculiar advantages and he did not agree that noise or vibration could be avoided by adopting special methods of design. It was perfectly true that gears which were designed without regard to kinematical considerations could not possibly operate successfully. Vibrations and noise in straight-tooth gears could only be avoided by careful design applied with full knowledge of limiting factors and with the most accurate workmanship obtainable. In some applications these requirements could not be met or applied, and the only satisfactory solution was the adoption of double helical gears.

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[1] Consulting Engr., Farrel Foundry & Machine Co., Buffalo, N. Y.

A. L. De Leeuw,[1] in discussing load between teeth in contact, thought that if there were any variation in the teeth, any slight inequality in the spacing, the load on a tooth might be very different from the theoretical load. Even if the variation were as low as 0.0001 in., perfect action might be prevented and the teeth might carry either an overload or only a partial load, and possibly no load at all. If the tooth proportions referred to by the author could be obtained with absolute accuracy, the action of the gear might possibly be all that he claimed for it. However, absolute accuracy was not obtainable, and in all the gears which Mr. De Leeuw had had a chance to observe, he had found slight but measurable inaccuracies. He believed that what was needed more than anything else at the present time was the limitation of such inaccuracies. Vibration also in toothed gearing deserved some real attention and was even now one of the most important problems met in machine-shop practice.

The author, in closing, replying to Mr. Sykes' first point, said that in so far as he was concerned personally, practically all of the work presented in the paper had been original, the references cited by Mr. Sykes being new to him.

Replying to Mr. Sykes' second point, gears might of course be designed to have not only a single pair of teeth continuously in contact but two or three or more. Replying to his third point, he assumed, as did also the author of What Makes a Gear Noisy? in the *American Machinist* for April 3, 1924, that it was a well-known fact among experienced gear men that great accuracy in cutting aligning, etc., would not always prevent noise in gearing. The reason for this had not hitherto been understood.

Mr. De Leeuw's and Mr. Sykes' objection that it was impracticable—not to say impossible—to cut gearing so accurately that just at the instant one pair of teeth of a pair of rotating gears was leaving contact a following pair would be coming into contact was most natural and obvious to an experienced engineer. Probably the best way to make clear the fallacy of this objection was to cite the case of an actual pair of ordinary double helical gears, which, *whatever their inaccuracies*, were quiet at full and overloads, and also quiet at no load, but as one certain intermediate value of the load was approached, became excessively noisy. This pointed at once to a resonant effect. Therefore, since the gears were quiet at non-resonant loads, all that was necessary in order to stop the vibration and howling was to *break up the regularity* of the frequency of the change in the number of load-carrying teeth and the job was done. In a pair of straight-face spur gears cut with ordinary accuracy but with the inaccuracies well distributed, and designed for an integral number of teeth in contact, the regularity of the frequency would be effectively broken up and with it, of course, the resonance which was the cause of the vibration and howling. For example, in a pair of gears designed for the integral number of two pairs of teeth in contact, this would result at the very worst in 1 tooth, 3 teeth, and 2 teeth in contact at *irregular* intervals, so that the resonance, and with it the noise, would be effectually prevented.

Compared with the straight-face spur gear, the helical gear has still less need of extraordinary accuracy. In order to understand why this was so, it must first be understood that the condition which it was desired to obtain in either the straight-face spur gear or the helical gear was not primarily an integral number of pairs of teeth in contact at all times, but really a constant total length of load-carrying tooth. In a straight-face spur gear designed for two teeth in contact and having a 6-in. width of face, the total length of load-carrying tooth was of course 12 in. In the helical gear we might disregard the "integral number of pairs of teeth in contact" condition and obtain a constant total length of load-carrying tooth by cutting the two halves of the gear in the proper relation to each other and giving them the proper width of face so that when a tooth on one gear half started to leave contact, a tooth on the other gear half would be completing leaving contact, and when a tooth began to enter contact, another would be completing entering, thus maintaining the total length of load-carrying tooth constant and preventing vibration of the gear and its load. Since on account of the angle of the tooth across the face of the gear a given tooth came into and left contact gradually, not all at once as in a straight-face spur gear, slight variations from the theoretically perfect affected the helical gear not at all. Since this method of dealing with helical gears involved no change in present tooth standards and no change in hobs or gear-cutting machinery, this was an important method of design. Helical gears not so designed might easily have a 100 per cent variation in total length of load-carrying tooth, considering the minimum length as the base. Worm gears were also subject to as large variation in the total length of load-carrying tooth.

In any type of gearing, when the condition of resonant vibration was encountered, the more accurate the cutting and aligning the sharper the "tuning," and consequently the greater the vibration and howling—not the reverse, as assumed by Mr. Sykes.

The variables which entered into the design of gears having a constant total length of load carrying tooth were: gear ratio, number of teeth, center distance, length of addendum, and pressure angle; and in helical gears, the helix angle and width of face. With so many variables the designs were very flexible and almost any special sets of conditions required could be easily met, including standardization.

[1] Consulting Engineer, New York City. Mem. A.S.M.E.

## Welfare Work of Sweden's Largest Industrial Company

STORA Kopparbergs Bergslags Aktiebolag (Big Copper Mountain's Mining Co.), abbreviated "Bergslaget," dating back to the thirteenth century, may claim to be the oldest industrial enterprise not only in Sweden but even in the whole world. The first object of its activity was the old copper mine at Falun, famous in the history of Sweden. Worked now for 700 years, this mine has for centuries played an exceptionally important part in the economic life of the country. Its most prosperous period was in the seventeenth century, when the company was the greatest copper producer in the world. After that time, however, its yield of copper gradually declined, although later on science and technics made it possible again to utilize the products of the mine, especially the pyrites, upon which an extensive chemical industry is now based, making the Falun Mine & Copper Works an important part of the concern.

The Bergslaget Technical School was established in 1908 by Dr. E. J. Ljungberg with a capital of 100,000 kroner, divided into ten shares. These shares carry no dividends, but all receipts of the school company are used for the promotion of technical instruction for members of the families of Bergslaget employees. The following year Dr. Ljungberg donated the main part of his fortune to this school company. Schools for instruction in rural household science and in housewifery were opened for girls. There are now 18 of these schools in operation: Six of these are vocational high schools, while three are schools for training in carpentry work and other mechanic arts, two are apprentice schools for industrial trades, two are schools for training mining foremen, and five schools for domestic science for girls.

Extensive measures have been taken for the social welfare of the company's employees. The majority of the workmen live in houses built and administered by the company. Furthermore, the company's stores at the different works supply provisions and other commodities at the lowest prices possible to obtain through wholesale purchases in Stockholm. The hotels, clubs, and canteens conducted at the various works also deserve mention.

For medical attendance and hygiene, the company has its doctors, nurses and hospitals, country homes for children, recreation homes and baths. Special welfare offices have been established for the care, wholly or in part, of the economic interests of the employees. All officials and foremen are granted pensions upon retirement and lately a pension has been arranged also for the common workmen. Every employee is insured against accidents.

The company's technic-scientific and cultural activity is very wide. In Falun there is a large chemical central laboratory and there are similar institutions connected with all the other works. There is also a large central library at Falun, well supplied with literature on all subjects included in the company's fields of activity with branch libraries at the various works. The company has a museum of industrial history, housed in an old administration building at the Falun mine, which is unique of its kind in Sweden. (From Sweden's Largest Industrial Company, by A. B. Sloane, in *Iron Age*, Sept. 25, 1924, pp. 751–754.)

# The Role of the Engineer in Industrial Mobilization Planning

By CAPT. E. E. MacMORLAND,[1] U. S. A.

*In order that the industrial mobilization of the country shall be prompt and effective in event of war, the Assistant Secretary of War has divided the country along lines of industrial density into fourteen procurement districts. In these districts, manufacturers who are to participate in the war program, are presented with drawings, specifications and information on the production of the items assigned to them so that upon orders from the War Department they can put their plants at the service of the Government. The author shows how engineering societies can aid the Government materially in its program by providing reserve officers, by giving advice on technical problems and assisting in the development of new weapons, by carrying the doctrine of industrial mobilization to the industries and developing their good-will, by assisting in the local plant surveys in areas which are not reached by the district personnel, by criticizing standard war contracts and by making special studies of materials, transportation, etc.*

THE CONDUCT of modern war is largely conditioned by munitions supply. It was frequently the boast of rival military commanders in the World War that all of their men were brave. There were few examples of panic-stricken routs of combat troops. Victory usually rested with the side having the heaviest fire power. The well-supplied German army in March 1918, broke its way through the Allied lines by superior munitions power. It failed in its triumphant march on Paris because its munitions power could not be sustained. The Russian hordes swept over the Austrian armies as long as their munitions supply was adequate. But brave soldiers were easily blasted back across the frontiers when the balance of munitions power passed to the Teutonic powers, and excellent battalions had only bare hands to resist a shower of iron.

It is just as true that munitions alone cannot win a war. Only men can clinch the victory. But munitions can lose a war if the troops fail to receive the supplies necessary to conduct combat. It is also true that wars can no longer be conducted by the armed forces alone. Every resource of the rival contenders must be thrown into the balance to make possible a bid for victory. Every dollar, every factory, every citizen must be enrolled in the cause. The nation with the larger resources and the better employment of its means will inevitably win.

The development of munitions power is an important factor in the bid for victory. It is not an extreme assertion to insist that it will always be a controlling factor. Men can be trained and put into the field long before the weapons and ammunition so necessary for victory can be produced for their use. It has been estimated that it requires a year longer to deploy munitions power than to raise and train men.

Munitions production has always been the task of the army of industry behind the army in the field. The natural line of cleavage between the army of industry and the army in the field is recognized in the National Defense Act of 1920. The army in the field is delegated under the provisions of this act for development to the military branches of the Army and that of the army of industry to the business head of the War Department, the Assistant Secretary of War.

The task of the Assistant Secretary is the more difficult of the two. He must arrange for the supply of munitions which are not usually in existence and require an appreciable time for fabrication. If he delays until war is declared in setting about the task, the field army will either have to enter the combat zone without munitions or not enter the field at all for ten or twelve months. The nation would not tolerate either condition, and it thus becomes necessary for the Assistant Secretary to take every step possible in time of peace to insure an early start in the munitions program in the event of another war.

## Plan for Mobilizing Industry

It is concerning the plans under way in the War Department for mobilizing industry and securing that early preponderance in munitions which is so necessary for a successful and early termination of a war of the future that I desire to speak. The plans of our own branch, the Ordnance Department, will form the principal theme of discussion, but it should be borne in mind, however, that other branches of the War Department are also active along similar lines under the supervision of the Assistant Secretary of War.

The basis for all industrial plans is found in the basic mobilization plan of the General Staff. This in turn depends upon an estimate of the situation covering information on possible enemies of the United States and the necessary organized man power to meet such enemies. The establishment of the basic mobilization plan, essentially a function of the General Staff, is the signal for the preparation of industrial plans.

The first step is to estimate the requirements of matériel month by month, to meet the mobilization plan of the General Staff. Each supply branch—Quartermaster, Ordnance, Signal Corps, etc.—is charged under the supervision of the Assistant Secretary of War with the provision of certain equipment for the army. The number of issue items so divided is in the neighborhood of 70,000. If components and raw materials to be procured were classed as items, the total would be nearer 500,000. It will thus be seen that the computation of the requirements by the branches is no easy task. In fact, it took fifty officers a year to calculate the essential items of this list and provide a basis for further work.

The scope and the extent of the procurement problem are indicated by the requirements. The next problem for solution is that of the sources of the production necessary to meet the requirements. It is here that the approach to industry begins. Where are we to get the munitions? The only answer is that commercial industry must provide the bulk of the production. It might be urged that the Government arsenals are maintained for this purpose. While the arsenals will in time of war produce a small proportion of the needs of the service, i.e., about 5 per cent, it is not as producing plants that they will be particularly valuable. They will be of more use as technical centers for the dissemination of information to industry in war time and as centers of the art of munitions making in time of peace.

## Decentralization of Planning Activities

Having found out what we want and that commercial industry is our practical source, how are we to establish contact with commercial industry and bring our plans to the sources? The answer is to decentralize the planning activities to areas where industry is located and institute active contacts with commercial producers. This is accomplished through the War Department district system. Fourteen War Department districts have been established in the various industrial areas and each supply branch is maintaining a field organization to keep in touch with industry. The Ordnance Department has established all fourteen of these offices. The personnel consists of only three persons in each—a district chief who is in all cases a prominent business man on a dollar-a-year basis, a regular army ordnance officer who handles the details of the work, and one civilian clerk. With this skeleton force we expect to go as far as we can in the making of plans for industrial mobilization for war. It may be that we shall have to increase our force later. If it becomes necessary, we shall be glad to divert more of the limited funds available to district work since it is felt that the importance of the problem of industrial mobilization warrants every effort we can put into it.

[1] Ordnance Department, Washington, D. C.

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### COÖPERATION OF INDUSTRY

Decentralization accomplished by means of the district system, it is now pertinent to the issue to examine the operation of the districts.

The total production which it is desired to obtain is determined in Washington after a study of the maximum monthly requirements of each item. This total production is then divided among the various districts in proportion to the supposed industrial capabilities of each. For example, the problem of securing 4,000,000 complete rounds of 75-mm. shell per month is divided among seven or eight districts. The districts selected are those whose industrial capacity is assumed to be most suitable for this production. Similarly a smaller procurement problem, such as that of the small 37-mm. shell, requirements about 600,000 a month, is divided among the three districts thought best suited for the purpose.

With the allocation, or "war order," are furnished drawings and specifications for the item. If available, a typical successful World-War method of manufacture is also furnished. Armed with this information, and under the supervision of the civilian district chief, the regular army officer in the district office then approaches the manufacturers thought to be best qualified by organization and equipment to undertake in time of war the fabrication of the item. If the manufacturer agrees to accept a war allocation after a study of the problem, the district chief then negotiates a standard contract, the forms of which are now in course of preparation, contingent on a war in the next three years and renewable every three years thereafter. It then remains for the contractor to make a plan for the utilization of his plant for immediate production in time of war when his contract is officially released. After placing the contract, it is the duty of the Government to keep the contractor supplied with the necessary information for his order and to take such other steps as may be desirable to keep up his coöperation and interest.

The plan in outline is simple, but its details are difficult of execution. It will be years before many details can be worked out. The success of the work will be directly conditioned by the response of industry. If we can secure the whole-hearted coöperation and good-will of industry, the path will be smoothed appreciably. Without such help we are wasting our time and the conclusion will have to be reached that our people do not want a relatively cheap form of preparedness. We are glad to say that so far the response has been favorable and that we feel that our plans will bear fruit if allowed the time necessary for full development.

### COÖPERATION OF ENGINEERING SOCIETIES

The engineering societies are needed in this work. The key men of the industries desired for war production are usually members of some engineering society. Particularly is the Ordnance Department in need of The American Society of Mechanical Engineers. In time of war we look to the Society for the best of our reserve officers to supervise war production; we expect your advice on technical problems; we believe that you can and will assist in the development of new weapons, which will exercise decisive effects on the outcome of the war. In time of peace you can be especially valuable in carrying the doctrine of industrial preparedness to the industries with which you are connected. You can lend a wholesome stimulus to the plant industrial plans which form the keystone of the arch of industrial mobilization.

We have made this appeal before and have obtained a gratifying response. Your society has pledged its aid and is assisting us through its Ordnance Division. The Iron and Steel Institute is actively engaged in studying the steel situation for us. Local chapters of engineering societies have endorsed the movement. Chambers of commerce and manufacturers' associations are smoothing the way for our field agents everywhere. The seed has been well sown; it now remains to be seen whether actual manufacturing plans will be the fruit of the plant so well nourished by the activity of the coöperating elements.

The solution of the problem of coöperation and good-will leaves us face to face with other more concrete propositions relating to industrial preparedness as a peace-time study.

### RESERVE OFFICERS NEEDED

We need trained personnel as Reserve Officers. In our districts practically all of the war-time production, inspection, shipment and payment activities will be in the hands of reserve officers. We need intelligent men for the handling of such a business. It runs into billions of the taxpayers' money. Disastrous delays in production must be prevented by these officers. Losses to the contractors and the Government must be minimized. The engineering societies can lend valuable assistance in helping us to secure the proper reserve officers for our war-time organization. Only the most capable men are wanted for such places; the responsibilities will be heavy and we want to place them in every instance on shoulders able to bear them. These officers can best be secured in time of peace when careful consideration can be given to the merits of the candidates. There is a considerable field for your society in putting a shoulder to the wheel in assisting in lining up personnel for our war organization.

The amount of money available for our studies in the field is limited. It can not be used in making plans for the manufacturer, i.e., we cannot let the manufacturer have an allotment of money to make plans for his plant. We must depend on his good-will to make these plans for himself, aided of course so far as is possible by the district office. Your society can assist us in getting the manufacturer to make adequate plans for the conversion and employment of his plant in time of war. There is no prospect of gain in these plans for the manufacturer. He has the sole satisfaction of knowing that he is contributing his bit to a cause which in the total will provide the United States with the best, most effective, and most economical form of preparedness. The measure of success in industrial planning will be directly indicated by the extent to which the individual manufacturers contribute in making plans for their plants. So far the response has been gratifying. One manufacturer in the Boston District has even gone further than planning. He has actually carried on some experimental tooling for his war order. Many others are collaborating with the district offices in working out plans as factory conditions will permit. Progress is being made, but we should like to see the work go even faster.

### TECHNICAL ASSISTANCE

There is a need for assistance along technical lines. For example, we should like to have some advice on our specifications. We are endeavoring to standardize specifications and to approach commercial standards as nearly as practicable. The closer we get to commercial standards, the better chance we have of getting the material in time of war. It is also our aim to stabilize these specifications and fight the war without changing them. There are sad precedents of costly delays in the World-War production due to specifications changes. With your advice we will have them correct at the outset and then stick to them throughout the emergency.

Similarly, we propose to standardize our types of material. Design changes wrought havoc with World-War production. It is our desire to settle on types which are producible in quantity and resist any changes for at least ten years. Some of you might assist us by examining some of these new types which will be made standard when fully tested, and give us your opinion on their producibility.

### PRESERVATION OF INDUSTRIAL METHODS

The war left us with a large group of commercial plants experienced in munitions production. The art was thoroughly mastered under the stress of circumstances and was well disseminated throughout industry. But the knowledge is disappearing. Commercial plants have returned to their peace-time activities; the records of the methods employed in munitions manufacture are fragmentary; the personnel who carried the burden during the war are scattering as time passes. The preservation of the methods employed by the most successful producers of munitions thus becomes an essential element in true preparedness. The Ordnance Department is engaged at present, through the medium of the districts, in gathering and collating such information. In final form, there will be a book covering a successful method of manufacture for each major item in the program for war procurement. It is felt that such books cannot fail to start the war contractor of ten or twenty years in the future on his production problem

with a minimum of delay and experimentation. The American Society of Mechanical Engineers might enter this problem by appointing a committee to criticize these books as they are prepared. It may be that the recent progress of engineering knowledge and practice will modify materially some of them. We do not contend that the methods of 1918 will necessarily be suitable for 1935. Perhaps they are out of date at the present time.

Certain commercial processes needed for ordnance production are steadily disappearing. Acid steel making is one of these processes. The Ordnance Department will of course take steps to preserve the art in the arsenals. However, metallurgical engineers could help us with studies showing the status of this process at present, steps to be taken to revive it in time of war, etc. This is a field where considerable assistance can be rendered by engineers more in touch with industry than is the Ordnance Department.

### STANDARDIZATION OF CONTRACTS

Standard contracts, aiming at the elimination of many of the contract difficulties which hindered World-War production, are in process in the War Department at present. The multitude of contract forms used during the war has been reduced to about five typical forms. The producers are being consulted in the formation of these contracts. Already many constructive suggestions have been received. However, in order to secure the best results in a war-time instrument, it is necessary to have the widest circulation possible. The engineers can assist in this work by offering their suggestions on the proposed forms. In short, we want to get a standard set of contracts which will satisfy everybody—a rather difficult task requiring as many heads thinking about it as possible.

### MATERIALS SOURCES AND TRANSPORTATION

Raw materials and transportation were found to be critical considerations during the World War. Raw materials were often lacking when most needed. Transportation difficulties frequently stopped production entirely in certain areas. A primary object in war planning is to place production of munitions near the largest supply of essential raw materials. Both factors are thus minimized. Studies of the sources and availability of raw materials, the best centers for munitions production, and the most efficient use of a transportation system normally taxed almost to capacity to move the peace-time production of the country, are important phases of our industrial war plans. We could use engineers for this work. Particularly, would it be important to know that we are not so overloading certain areas with war orders that it would be impossible to move raw materials into the plants and the finished articles out of them.

### INDUSTRIAL SURVEYS

Occasionally we have need of a friend for plant-survey work in distant areas where our district personnel cannot make a personal visit. It would be of great assistance if members of the Society would list themselves with the nearest district office of the Ordnance Department if willing to spend a little time in local industrial surveys. We do not send out questionnaires, all contacts being personal. Some of the districts are very large and it is practically impossible for the Regular Army officer in the district to get over the ground completely. We rely on reserve officers and other friends to assist us in distant areas.

I have sketched hurriedly the problem and a few of the ways in which The American Society of Mechanical Engineers may assist. The Society as a whole has come out for coöperation and is active on certain problems for us at present. We want to line up every member in the cause if for no other reason than that of developing good-will for us in the organizations with which they are connected. Much can be done with good-will as a basis, and little without it.

Now what will be the effect if this problem is solved and the country stands in a true position of industrial preparedness? By such plans, we have at relatively small expense bought an insurance policy which may prevent entirely the occurrence of war. No other power will, it is safe to assume, attack us if we have made plans to provide a preponderance of munitions in the shortest possible time. Victory is impossible against a superior weight of munitions. While the prevention of war by adequate preparedness is the primary goal of the War Department, it is realized that war may be thrust upon us anyway. If such is the case, early preponderance of munitions will surely operate to shorten the war. Since wars are tremendously expensive in both men and money, every hour cut from the end of the conflict represents a huge saving. Consequently whatever happens in the future, planning for industrial mobilization cannot fail to be worthy of the time, money, and effort put into the work. In the last analysis, it will provide at little expense a form of preparedness against war or in case of war which has never before been approached in the history of the United States.

## Discussion

CAPTAIN MACMORLAND'S paper contributed by the Ordnance Division and presented at the Spring Meeting of The American Society of Mechanical Engineers, Cleveland, Ohio, on May 28, 1924, with Major Fred J. Miller presiding, was followed mainly by oral discussion.

Howard Bruce[1] thought that the most important thing in the preparedness program was to retain the knowledge of production methods obtained during the war, by reduction of this knowledge to actual paper record. Another important point he thought to be the question of specifications, which should be placed in the hands of manufacturers and kept up to date. Colonel Ledyard Cogswell,[2] speaking as a district chief, said that the A.S.M.E. could help in the work of industrial mobilization not only by supplying reserve officers but also by giving names of men in commercial life who could assist in many ways. R. M. Dravo[3] spoke of the almost unanimous accord of Pittsburgh manufacturers to help the Ordnance Department, and of the great success of propaganda meetings in that city. Colonel B. A. Franklin[4] gave a similar report of the feeling among manufacturers and engineers in Bridgeport, and urged the Society to take an active rather than a passive part in the work of preparedness. Colonel C. L. Harrison[5] thought the Society could help in securing inspectors in case of another emergency, and that engineers with their technical knowledge should coöperate with accountants. J. C. Jones[6] emphasized the extent to which the Ordnance Department was dependent on the engineering profession, and in view of the enormous number of articles required in war time and of the complexity of specifications, thought that above all engineers should be well informed on matters pertaining to industrial mobilization. Colonel Bascom Little[7] described the wholehearted way in which the Cleveland Section of the A.S.M.E. had coöperated with the Ordnance District authorities. Liaison between manufacturers and the Government has been carefully worked out. When a war order is assigned to a plant, a production officer in its organization is selected and asked to take a commission in the Reserve Corps. The local section will be asked to nominate the equally important inspectors. A. E. Russell[8] suggested that those men who did not care to join the reserve at this time might join the Army Ordnance Association, and in this way keep in touch with the Ordnance Department.

Gerald Cahill[9] submitted a written discussion, an abstract of which follows.

Captain MacMorland states that the number of issue items is in the neighborhood of 70,000. Chief among these items is ammunition of various calibers, from that of the 14-in. gun and its high-explosive shell down to the 45-caliber automatic pistol and the 30-caliber Springfield rifle and their respective cartridges. The 30-caliber ball cartridge is the principal one of the major munition items. It is used in the magazine rifle, automatic rifle, machine gun for infantry, and the machine gun for airplane equipment.

During the 19 months we were at war, approximately 3,500,000,000 rounds of ammunition were manufactured, and of this quantity 2,800,000,000 rounds were 30-caliber ball cartridges. When the

[1] District Ordnance Chief, Baltimore, Md.
[2] District Ordnance Chief, Buffalo, N. Y.
[3] District Ordnance Chief, Pittsburgh, Pa.
[4] District Ordnance Chief, Bridgeport, Conn. Mem. A.S.M.E.
[5] District Ordnance Chief, Cincinnati, Ohio.
[6] District Ordnance Chief, Philadelphia, Pa.
[7] District Ordnance Chief, Cleveland, Ohio.
[8] District Ordnance Chief, Chicago, Ill.
[9] Mgr., Lowell Works, United States Cartridge Co., Lowell, Mass.

Armistice was signed the cartridge companies were shipping approximately 12,000,000 rounds of cartridges every 24 hours.

The initial delivery of small-arms cartridges began within two months after we entered the war at the rate of approximately 2,000,000 cartridges per day, and in seven months after we entered the war there had been delivered approximately 150,000,000 cartridges.

This was a record for rapid production, but it gives us a wrong and dangerous conception of the length of time required for the mobilization of the sporting-cartridge industry for the manufacture of military cartridges in case of another war.

We must not neglect the fact that when we entered the war the five principal cartridge companies in the United States had been two and one-half years manufacturing military cartridges in large quantities and that they had expanded their plants to abnormal capacities to fill the urgent requirements of the European governments. So that when we entered the war the cartridge-manufacturing companies had available for our Government an abnormal installation of military-cartridge-making equipment with an estimated capacity of 5,000,000 cartridges per 24-hr. day, trained personnel and skilled operatives, and it required but a very short time to change over their equipment for the manufacture of our 30-caliber cartridge.

The conditions existing in this country in respect to the manufacture of military cartridges at the time the European or World War began will be analogous to those which would occur should we enter another war—unless the program for the mobilization of industry is carried on to a successful conclusion. To illustrate more clearly the time required to get into mass production of military cartridges, the experience of the United States Cartridge Co., may be cited.

At the time the European conflict began they had an equipment for the manufacture of 70,000 military cartridges per day. When contracts were received steps were immediately taken to expand and equip their plant for the manufacture of 2,000,000 military cartridges per 24-hr. day. The machinery for the manufacture of sporting ammunition is in but very few cases suitable for the manufacture of military cartridges. Cartridge-manufacturing machinery being of special design, it is not carried in stock but built only on order. With machine builders and our personnel working under great stress and an excellent intelligent-labor market from which to select mechanics and operatives, it nevertheless required six months to attain a production of 350,000 military cartridges per day and one year to attain a production of 1,000,000 per day. The total accumulative production of the five cartridge companies one year after the receipt of European military-cartridge orders was less than 100,000,000 cartridges—Frankford Arsenal's one-year peace-time capacity. Compare this with the quantity delivered to our Government one year after we entered the war—1,318,000,000. It is not a comparison, it is a contrast, and impresses upon us the importance of intelligent industrial mobilization of the cartridge industry.

The 30-caliber military cartridge differs from other major munition items in the fact that nearly all of its 35 machine operations are performed on special-design machines. To build a unit for the manufacture of 100,000 military cartridges per day of twenty-four hours will require approximately 20,000 man-hours, so we can appreciate the length of time required to build sufficient machinery to meet the urgent requirements of war.

The large quantity production made necessary by the World War lead to the adoption of a new method of cartridge-case manufacture. This was known as the Hooker extrusion process. It was developed by the U. S. Cartridge Co., and at the signing of the Armistice they were manufacturing by this process 2,000,000 cartridges per day. To install a manufacturing unit of this process for the production of 100,000 cartridges per day would require about 17,000 man-hours, so that developments along these lines of cartridge manufacture would be of considerable importance in case of another war emergency.

The A.S.M.E. could recommend as reserve officers certain members who have special training along lines allied to cartridge manufacture, who should be assigned at various times in the year to the Government cartridge plant at Frankford Arsenal where they could become familiar with the manufacturing details of cartridge manufacture. Their training should be along the fundamentals of cartridge manufacture, such as tool design, the proper working or displacement of metal, and also ballistics. It would be advisable to specialize on one component of the cartridge, so that in the event of another war they would be competent to instruct and train key men in the art.

Cartridge manufacture is a highly specialized product and it takes practice to attain its technique. Unless something along these lines is done there will not be a sufficient number of technical men skilled in the difficult art of military-cartridge manufacture to meet the urgent requirements for the military cartridges necessary in case of war.

# Recent Developments in Electric Locomotives

TWO papers dealing with recent developments in electric locomotives were read at one of the sessions of the A.S.M.E. Spring Meeting in Cleveland. The first, by N. W. Storer, of the Westinghouse Elec. & Mfg. Co., gave tabulated data of types, dimensions, weights, capacities, etc. for a number of recent designs and set forth the reasons why the types discussed had been developed. The second paper, by A. H. Armstrong, of the General Electric Co., gave a general comparison of steam and electric locomotives, discussed at length the four forms of motor drive used and the factors affecting the determination of tractive-effort rating, and gave particulars regarding the otheograph and its use in measuring vertical and lateral impact on rails resulting from the passage of locomotives or other rolling stock. Mr. Storer's paper was published in Mechanical Engineering for September, page 523; Mr. Armstrong's appeared in the October issue, page 608. An abridgment of the discussion elicited at the session is given below.

Edwin B. Katte.[1] Electric locomotives have been in successful service on steam railroads for over twenty years and it would seem that by now the manufacturers should have acquired sufficient experience to enable them to determine some of the generic principles involved and to concentrate on a few common types of electric locomotives and obviate the enormous development charges habitually tacked on to every proposal submitted for electric locomotives.

[1] Chief Engineer, Elec. Traction, N. Y. C. R. R., New York, N. Y. Mem. A.S.M.E.

In any electrification project not involving suburban service the cost of the electric locomotives is the major portion of the investment involved. This cost within the past few years has mounted from about 25 cents per lb. on drivers to about 85 cents per lb., the latter price including development charges. The most recent quotations which the writer has seen for heavy freight electric locomotives in lots of thirty or more was about 55 cents per lb. on drivers, whereas Mikado-type steam locomotives were this year purchased for 32.6 cents per lb. on drivers.

Again, if these quotations are reduced to cost per 1000 lb. tractive effort, we find that the price quoted for electric locomotives increased in one year from $2546 per 1000 lb. tractive effort to a minimum of $3704 and as high as $5145 in 1924, whereas the Mikado-type locomotive in 1923 cost $2026 per 1000 lb. tractive effort and this year similar locomotives have been bought on the basis of $2132 per 1000 lb. tractive effort.

The writer believes he is correct in saying that the cost of electric locomotives has increased more than any other commodity required by steam railroads, and in his opinion the lack of stability in the cost of electric locomotives has done more to retard the electrification of steam railroads than any other thing, or perhaps all other things combined. Electrification of steam railroads cannot become general until the manufacturers concentrate upon a few general types of electric locomotives.

Up to the present time very little information has been made available with regard to the probable life of electric locomotives.

The first 35 locomotives purchased by the New York Central Railroad have been in passenger-car switching service for 17 years and are now about to be shopped for general overhauling. With steam locomotives in about the same switching service, it is the practice to shop them every five years.

It has been calculated that the cost of rehabilitating these electric locomotives will average about $12,000 each, and that after such work has been performed the locomotives will be practically as good as new and their useful life will be extended from 15 to 20 years; thus the total life of these locomotives may be expected to be between 35 and 40 years.

A new locomotive of the same weight and capacity to replace an old locomotive now due for general overhauling might cost between $90,000 and $110,000. This is based upon a unit price of from 40 to 50 cents per lb. Hence the wisdom of repairing and renewing worn parts of the old locomotives is clearly apparent under the present market conditions.

E. WANAMAKER.[1] The writer is primarily interested in electrification developments such as are applicable to (comparatively speaking) lines of lighter traffic.

The possibilities of the design incorporated in the Ford D. & I. locomotive promises much for such lighter-traffic lines.

With trains comparatively infrequent, power stations far apart, etc., the moving substation or the combined locomotive and substation, as it may be called, may easily prove the ultimate solution.

It is apparent that on lines of lighter traffic, a locomotive (and this is especially true on prairie lines operating out of such a railroad city as Chicago) such as the Ford D. & I. design could easily prove of great economic value. This is also especially true where there is considerable fluctuation in the traffic. To secure net operating economy, the detailed design of these locomotives should be such as would enable the operating forces to keep such locomotives in service as nearly as possible 100 per cent of the time.

Mr. Storer's paper brings out (between the lines) the desirability of the development and use of internal-combustion-engine-drive motor cars and oil-electric locomotives for non-electrified branch-line service and for switching service in electrified zones. The use of such equipment tends to reduce the first cost of electrification and at the same time betters the service and reduces the cost of operation.

R. EKSERGIAN.[2] Strange as it may seem, the electric side has proven far more positive than the mechanical design, so much so in fact that it might be now safely stated that the mechanical design offers the greatest obstacle in the future development of electric locomotives and in no small measure to the actual success of electrification. This difficulty is due to the peculiar conditions of establishing an ideal drive and meeting the present limitations imposed by the proportions and designs of electric equipment now used, combined with good tracking qualities, proper weight distribution, reliability of mechanical strength and low maintenance.

The axle-hung-motor drive is undoubtedly the most simple and rugged for slow-speed service. Due to the relatively high r.p.m. obtained by the gear ratio a simple and compact motor is possible. For high speed this type is not adaptable since with the lower gear ratio needed, and the consequent higher torque demand on the motor at low speeds, the motor-frame size becomes excessive for the limited space between the drivers in large-size locomotives. In addition the unsprung weight, tooth stresses at high speeds, etc., offer serious difficulties for high speeds. Even for freight locomotives of large capacity the axle-hung-motor design is limited due to space available for motors of large capacity and the spreading of the rigid wheelbase, which offers difficulty in tracking.

The direct or bipolar drive is adaptable primarily for high-speed service. The serious objection to this drive is that the armature is entirely unsprung. This requires a very rugged armature and limits the allowable axle load to a considerably lower value than for other types. The lower center of gravity is also an objectionable feature in increasing lateral impact stresses, though with the introduction of spirals, etc., this feature has been greatly overestimated. Its great advantage, however, is in the compact wheelbase possible and the general simplicity of the design.

[1] Electrical Engineer, Rock Island Lines, Chicago, Ill.

[2] Engineer, The Baldwin Locomotive Works, Philadelphia, Pa. Mem. A.S.M.E.

The quill drive seemingly offers one of the most ideal types of drives due to spring suspension of the motors, raising the center of gravity, etc. Difficulties are found in the detail design of the helical quill springs, in the interference of the quill spring system with the equalization on starting when the torque reaction on the motors is a maximum, and, constructively, in the spreading of the wheelbase. Spring difficulties have been overcome by housing the quill springs, but this necessitates a large driver, with consequent spreading of the wheelbase and difficulties in the design of a suitable wheel center due to the wide spacing of the spokes.

The side-rod drive is an approach toward steam practice, but the conditions of side-rod loading, however, are considerably different. In the first place, the change-over positions are more indefinite due to the necessary reduction in pin plays, etc., with consequent increases in rod loadings. Due to the large armature inertia effect increased by the square of the gear ratio, small rod plays are absolutely necessary to prevent large impact loadings on the rod, though this situation has been greatly relieved by the introduction of torsional flexibility at the jackshaft. Due to the decrease in allowable play as compared with steam practice, careful alignment is of extreme importance, together with the maintaining of accurate bearing centers. Therefore the frame design must be extremely rugged between the jackshaft and main driver journal. Another disadvantage of side-rod drive is the necessary grouping of motors at the jackshaft, usually at the ends of the locomotive, and the consequent spreading of the guiding truck from the rigid wheelbase, thus introducing an excessive overhang and necessitating as well the use of only a two-wheel truck. The great advantage of side-rod drive is in ample motor space, the possibility if desired of using large motors with ample capacity and in the compact grouping, allowing a reduction in the length of rigid wheelbase. Moreover the drive and motors are spring suspended and the overall weight efficiency is better than for any of the other types.

SIDNEY WITHINGTON.[1] Enough information should be available, if properly correlated among the interested railroads and manufacturers, to indicate the general arrangement of parts which is most economical for any given service, all things considered. It is difficult to escape the feeling that there may be an element of prejudice in the adoption by one railroad of jackshaft and side rod, by another of twin motors geared, and by another of gearless motors for virtually the same class of service. Steam-locomotive design has become standardized to a high degree, and it is time serious consideration were given to getting together coöperative information with a view toward standardization of electric locomotives.

The performance of the original New Haven locomotives, in which the motor is mounted on the axles and supported by quill springs, has been impressive. These locomotives are still operating satisfactorily in spite of more than a million miles apiece to their credit, and in spite of the fact that train weights have more than doubled since they were put into service, and the loads consequently are in excess of those they were designed to handle. The theory on which these locomotives were designed was the handling of light trains by a single unit, double- or triple-heading the heavier trains with locomotives operated as multiple units. This presents some distinct advantages in the matching of motive-power capacity to the train loads and avoiding unnecessary operation of surplus capacity on light trains. Another advantage is that when a unit is in the shop a minimum amount of capital is tied up.

This opportunity is taken to suggest a change in our means of designating electric-locomotive wheel arrangement, differentiating the driving wheels from ponies or trailer wheels. This might be done (as is the practice abroad) by assigning letters instead of numbers to the driving axles. Thus 2–4 + 4–2 would become 1–B + B–1 (counting axles instead of wheels) or 2–D + D–2 (counting wheels as we do in this country). The letters in the designation would have the added advantage of indicating in themselves that reference is made to an electric and not a steam locomotive. Some such formula will greatly simplify designation of motive power as time goes on and electric locomotives become more common. Cannot the appropriate A.S.M.E. Committee look into this?

[1] Electrical Engineer, N. Y. N. H. & H. R. R. Co., New Haven, Conn. Mem. A.S.M.E.

### Oral Discussion

James Partington,[1] who presided over the session, said that he believed all engineers would agree that the very extreme axle loadings employed on certain types of electric locomotives and steam locomotives were beyond the recommendations of the engineering profession. The desire of the railroads for extreme power was, he thought, largely responsible for these very heavy axle loadings.

N. W. Storer,[2] author of the first paper, said in closing that if Mr. Katte would refer to the concluding paragraph of his paper he would see that it stated that the tendency in heavy locomotive design for freight service seemed to be toward the use of heavy axle loadings and transmission by gears, jackshaft and side rod. That statement was based on the fact that the last three large locomotives sold in this country had that type of drive. However, he would be the last man to question the possibilities of other types, for instance, axle-hung motors. Mr. Storer agreed with Chairman Partington that the tendency was toward excessive weights on driving axles. Railway men always desired to get the maximum possible loading, but it was not at all a necessity in the case of the electric locomotive where the power could be so readily subdivided. He also agreed with Mr. Armstrong that the locomotive ought not to cause any more distress to the track than the train that followed it. As to Mr. Katte's statement regarding the number of types, this might be altogether the fault of the manufacturers, but he did not believe it was. Some of the latest locomotives that had been built had been dictated by the railways themselves or by their consulting engineers.

A. H. Armstrong,[1] who presented the second paper of the session, said that if he were to make any statement as to form of construction, it would be that for the switching locomotive the five-axle truck with axle-hung geared motor drive was almost universally accepted. For freight work he saw no reason at all why an axle-hung geared motor would not meet any problem put before the electrical engineer. For high-speed passenger service either the axle-hung gear-driven motor or the gearless drive was used, and there was something to be said for each. The problem was very much confused by bringing into such a simple thing as translating the power of a motor to the driving axle the question of the type of motor to use. The individual rod and quill drive had been necessary in order to apply alternating current to the heavy locomotive. The same forms of mechanical construction might or might not have been considered with direct current, but they would not have been arbitrarily imposed as was the case with alternating-current motors.

# Fourth New Haven Machine-Tool Exhibition

## Conducted under the Auspices of the A.S.M.E. New Haven Section, Yale University, and New Haven Chamber of Commerce, Proves Unqualified Success

THE Fourth Machine-Tool Exhibition held in New Haven from September 15 through 18 has been widely heralded as an unqualified success. The more than 15,000 visitors who came to New Haven for the show and the excellent technical sessions which accompanied it, taxed the hotels to their capacity, and many who had neglected to make advance reservations were forced to find refuge in adjacent cities. They came from twenty states and four foreign countries, and testified to the interest and value of the meetings and exhibit.

The Exhibition was conducted under the joint auspices of the New Haven Section of The American Society of Mechanical Engineers, Yale University, and the New Haven Chamber of Commerce. The Machine Shop Practice Division of The American Society of Mechanical Engineers, with the assistance of the Committee on Education and Training for the Industries, contributed the technical program. The Committee in charge was made up of H. R. Westcott, Chairman, E. Hartford, S. W. Dudley, W. W. Gaylord, J. D. Marsh, K. F. Lees, E. Buckingham, N. Horn, R. Oliver, R. McArthur, A. F. Breitenstein, E. B. Crawford, J. S. String, Wm. Buxbaum, and G. P. Simpson.

The Exhibition, which filled three floors of Mason Laboratory—the mechanical department of Sheffield Scientific School—was made up of showings of over 125 manufacturers, and many new machines and novel machine-shop devices were on view. The showings of small tools, gages, and collapsible taps and dies were especially attractive. The machine tools included all types, many of them in operation. The exhibits of various types of transmission devices, ball bearings, etc., were unusually complete. While the manufacturers of New England were well represented, there were many exhibitors from the Middle West, and the entire show gave an excellent cross-section of American machine-tool practice.

The technical program consisted of five sessions, two informal dinner conferences, and several plant visits. The opening session was held in Mason Laboratory and the others in Dunham Laboratory. The papers were presented before excellent audiences which indicated that this revival of A.S.M.E. activity in the machine-shop field was thoroughly appreciated.

The Exhibition also furnished the background for a number of important committee meetings. The work of standardizing tool holders and tool-post openings was initiated at an organization conference of the sub-committee in charge of this project. The Working Committee on Standardization of Tee Slots held a meeting at which it adopted a progress report. These two projects are under the supervision of the Sectional Committee on Small Tools and Machine-Tool Elements, which also met during the exhibition. The Sub-Committees on Methods of Gaging Manufactured Material, and on Gages, Their Limits, Manufacture, and Use, met jointly. These sub-committees are part of the Sectional Committee on the Standardization of Plain Limit Gages for General Engineering Work. All of the foregoing committees have been organized and are functioning under the procedure of the American Engineering Standards Committee (A.E.S.C.) In addition the Special A.S.M.E. Research Committee on Cutting and Forming Metals met, and the A.S.M.E. Committee on Meetings and Program held its regular monthly meeting.

The opening session, on Monday evening, September 15, which was reported in Mechanical Engineering for October, was addressed by Dr. George K. Burgess, Director of the U. S. Bureau of Standards, an abridgment of his remarks also appearing in that issue.

The first technical session was held on Tuesday afternoon, September 16, with Earle Buckingham in the chair, at which a paper entitled a Review of Foreign Progress in Research in Cutting Metals was presented by C. A. Beckett. This paper was published in the October issue of Mechanical Engineering. In the discussion B. H. Blood, Chairman of the Special Research Committee on Cutting and Forming Metals, pointed out that in the cutting of metals a large amount of research was necessary, the results of which should be available to all manufacturers and engineers interested. The problems to be solved were so far-reaching that their solution would require whole-hearted coöperation and assistance. Wilfred Lewis, of the Special Research Committee on Gears, described the testing machine which will be set up at the Massachusetts Institute of Technology and used experimentally to determine the effect of speed upon the strength of gear teeth and to secure data on the problems relating to gears, such as the frictional loss in the transmission of power and the conditions affecting noise. The second paper to be presented

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[2] General Engineer, Westinghouse Elec. & Mfg. Co., East Pittsburgh, Pa. Mem. A.S.M.E.

[1] Chairman Electrification Committee, General Electric Co., Schenectady, N. Y.

was one by Ernest F. DuBrul, in which he discussed the problem of forecasting the demand for machine tools. This paper appeared in the September issue of MECHANICAL ENGINEERING.

On Tuesday evening S. S. Edmands, Director of the School of Science and Technology, Pratt Institute, Brooklyn, N. Y., presided at the session devoted to Education and Training for the Industries. Mr. Edmands pointed out that the need for industrial education is real, that it is growing constantly from year to year, and that it is urgent. The trend of our immigration situation is to reduce the easy supply of foreign trained men. This makes it increasingly necessary to develop efficient industrial education in this country adequate to our needs.

The speaker of the evening was R. O. Small, of the Division of Vocational Training of the Massachusetts State Board of Education, Boston. Mr. Small emphasized the fact that an adequate scheme of public education should provide not only for the civic training of the individual, but for economic training, recognizing that the individual must earn a living. The prosperity of New England is dependent upon the prosperity of the industries of New England, and they will continue to be prosperous as long as they can secure a supply of properly skilled help from the operator up to the most highly trained technician. It must become the business of industry, not only through the coöperative school and the apprenticeship training carried on by industrial concerns, but through all the agencies, no matter where they may be found, to adequately train a selected group of individuals to earn a living.

What are the public-school agencies to which we can resort for giving this kind of training? "Manual training" is not going to fit boys or girls for placement in industry. The day school can be attended only by those who can forego earning. The brains of this country are just as likely to be found in an alley as on an avenue, and as it is necessary to capitalize all those resources, we must have some other device than the day school. Evening schools have done a great deal, but the peculiar thing about them is, that during periods of business depression, when individuals have a good deal of time on their hands, the evening schools are not filled. Part-time education would therefore seem to be indicated as the answer.

Industry is to be called upon to do a very much larger share of the work if we develop adequate part-time schemes of education, than they seem to appreciate is necessary at this time. The part-time or coöperative part-time school is one solution to the problem. A good illustration is the Beverly School of the United Shoe Machinery Company, which has stood the test for 14 years. The scheme of industrial training roughly is this: They have two groups of boys at the factory earning a wage, and another group at the school studying about the technical and related work which has to do with the machinist's trade. Each week the group in the school is moved into the shop and one of the groups in the shop into the school room. In this way the boys learn the trade as they work. Some of the boys are now assistant superintendents in the United Shoe Machinery plant.

A great deal has been done in the public schools during the last few years as regards preaching the dignity of labor, but many still have the old Greek notion that there cannot be much between the individual who works and the really educated gentleman. Training the youth of this nation for honest employment is a training of character. Unless the individual does something he does not develop character and culture.

A. C. Jewett, of the National Industrial Conference Board, New York City, presented a discussion in which he pointed out that education for the industries is not essentially or basically different from education for any other useful activity of life. The educational institution must develop correct thinking, it must develop ability to render sound judgment, it must develop personality. Mr. Jewett told of the study made by the National Industrial Conference Board to determine the relation between engineering education and industry with the idea of setting up specifications for a trained engineer required by industry. The result of this was the recommendation from manufacturers that engineering graduates be trained to think for themselves, and be well versed in economics and general subjects.

Frank Cushman, Chief of the Industrial Education Service of the Federal Board of Vocational Education, pointed out that for the year 1923 there were 325,000 persons enrolled in courses established in 48 states in coöperation with trade and industrial schools. During the last fiscal year the various states spent approximately $2.50 for vocational education for every dollar the Federal Government put into the work.

At the Wednesday afternoon session, over which Luther D. Burlingame presided, there were three papers presented, the first, by D. M. Chason, on Comparative Methods of Tool Design and Their Relation to Quantity Production of Sheet-Metal Parts, and the second, by Earle Buckingham, on Shop Measurements. These two papers appeared in the September issue of MECHANICAL ENGINEERING. The Application of Ball Bearings to Machine Tools was the subject of the third paper, by H. R. Reynolds, who stated the principles that must be followed for the successful use of ball bearings. He emphasized proper closure to retain lubrication, and proper size selection.

The final technical session was held Thursday afternoon under the chairmanship of Dr. George K. Burgess, Director of the Bureau of Standards. J. D. Pedersen, of the Springfield Armory, discussed the Manufacturing Design of Ordnance Matériel, and his paper will appear in abridged form in the December issue of MECHANICAL ENGINEERING. The remainder of the session was devoted to the subject of standardization and there was a wealth of interesting discussion. L. D. Burlingame's paper on Standardization Versus Individuality which appeared in the September MECHANICAL ENGINEERING attracted considerable attention. In it the author pointed out that standardization in the machine-tool industry is confronted with the need for a decision between two courses: efforts to improve and excel by novelty in design on the one hand, and standardization on the other. He stated the standardization projects at present under consideration, and extended a plea for a conservative policy rather than a headlong plunge into an overstandardization which would hamper progress.

Harry E. Harris, of the Harris Engineering Company, Bridgeport, presented a written discussion in which he took issue with Mr. Burlingame's plea for a conservative policy of standardization, pointing out that rapid progress in machine-tool development can be best attained by the aid of standardization of fundamental details. He also emphasized the fact that standardization projects, as carried out under the American Engineering Standards Committee, cannot be pushed too speedily as all phases of thought are represented on the committees and substantial agreement must be attained before any standards can be adopted. He also stated strongly that it was his opinion that there is no danger of overstandardization, because so far it has been difficult to find sufficient engineers who could devote the large amount of time necessary for standardization work. Mr. Harris pointed out as an example illustrating the fact that individuality hampers progress, the present tendency toward greater use of electric motors for individual drives on machine tools. Lack of standardization of revolutions per minute, motor-shaft diameters, and motor dimensions causes extra expense and costly delay to machine-tool builders as a well as users. A large number of varieties of motors may be used in one shop, causing a tremendous amount of delay for replacement and of increase in the investment for repair parts.

Arthur E. Norton, assistant professor of mechanical engineering Harvard University, referred to the important project of standardizing machine elements, such as bolts, nuts, rivets, screw threads, gears, shafting, etc., which is being carried on under the procedure of the American Engineering Standards Committee. He described the procedure, which provides that each standard be first studied and enacted by a competent committee; second, reviewed by the technical society or societies who sponsored it; and third, adopted by the representative American engineering standard committee who announces it as the American standard. He then went on to relate the progress being attained in the standardization of bolts, nuts, and rivets, of which committee he is chairman, and discussed the economies of standardizing such basic machine details.

It was generally accepted, he said, that from an economic standpoint such standardization should make machines cost less due to the reduction of the numbers of *styles* and possibly *sizes* of machine parts which must be made and kept in stock. In his opinion labor costs would not recede, and the only field for economic effort was that of simplification and standardization of parts. But while

we were doing this, we should be improving the product, which was, after all, just as important as lowering the cost.

This possibility of improving the product through standardization had been borne in on him as he had watched the men working on the committee.

From the point of view of an educator he was continually seeing the necessity of getting basic things settled, so as to be freed from the necessity of discussing the elementary forms and to have time and energy for studying new combinations for new needs. But basic things were not settled until they were settled right and he had come to believe that standardization of machine parts was the best way to start settling basic things for making better machines. Take for example the matter of bolts and nuts, now being studied by the sectional committee. The attempt to standardize these basic elements had immediately created discussion of what was the best form for the purpose. Experiment had been resorted to for final answer to the question of minimum thickness of a nut in certain cases. Clearances between wrench and nut had been studied and an interesting mathematical analysis had been made of tolerances to be allowed on both nut and wrench to avoid destruction of corners of the nut and too great turning angle for the wrench. Methods of production had been subjected to question and commercial tolerances analyzed.

Users of bolts and nuts often demanded of bolt manufacturers certain special refinements of form, not knowing about the standardized forms. Perhaps the manufacturers themselves did not know of any good reason why existing forms had come into being. This led to the conviction that any standardizing work should be national and also continuous. A body should be permanently available to explain and modify as well as to create standards.

If this was done, then a standard of *form* might become a record of the best way to make a basic machine part. If new processes or needs arose, new forms could be listed as special (and thus be subject to the higher cost at first) until the permanent standardizing body was convinced that the improvement was such as to warrant the modification of the . . . . . standards or the recognition of new forms in the family of standards. Thus it was necessary to educate the user, as well as enact standards. We must accompany all standards with well-edited literature giving some idea of the reasons for existing forms, and we must make standardizing work continuous, the standardizing bodies being always available for reference in case of dispute or protest.

Sometimes charges were brought against the whole theory of standardizing on the ground that it might put fetters on initiative and stifle thought. However, sound methods would prevent this. It was only desired to standardize parts which were basic; then engineers would be more free to devote their minds and energy to the greater problem of newer and better combinations in new and better machines.

Arthur M. Houser, of the Crane Company, Chicago, agreed with Mr. Burlingame that care must be used in standardization policies. Standardization appeared to him as a serious matter and he pleaded for a conservative line of standardization projects.

In his paper on Selling Standards to Manufacturing Organizations, Col. E. C. Peck, Chairman of the Standardization Committee of the A.S.M.E., pointed out that the surface of standardization had only been scratched and there was still a tremendous amount of work to be done if the United States was to take the position to which her previous work entitled her. He pointed out that progress had been made in developing working standards and that there were many committees busy under the procedure of the American Engineering Standards Committee. The machinery for setting up and developing a standard was adequate but at the present time there was no adequate means of getting these standards adopted by the industries to which they applied. Colonel Peck stated that it was his opinion, based on observation and constant inquiry, that standards were being adopted very slowly and were in many cases unknown. He pointed out that this condition could be due to one or more of three things: lack of information in the industry that there was a standard applying to that industry; lack of merit in the standard; and lack of interest due to the inertia of the engineering personnel of the industry concerned. He swept aside the objection of lack of merit in the standard by reviewing the careful procedure which must be followed before a standard can be approved by the American Engineering Standards Committee. He stated that favorable reports had been received from industries that had adopted the approved standards. He then went on to stress the effect of inertia on the part of the engineering profession and assuming that it was informed about standards that were available. He closed with a plea for more thorough study of this subject by engineers and for greater efforts to disseminate information about approved standards. He suggested that local engineering societies, which are active throughout the country, might well contribute a great deal to education in standardization.

George T. Trundle, of the George T. Trundle Co., Cleveland, discussed the value of standardization. He related a number of actual experiences covering several lines of industry in which he showed the value of standardization as measured by comparing a standardized condition with the former non-standardized situation. He related how one organization had purchased many different types of equipment for the same operation and indicated that large savings had been made by standardizing this equipment later when competitive conditions prevailed, with a great reduction in money tied up in storage of the different spare parts. In another instance there were no two lineshafts of the same diameter in the plant nor were the lineshaft speeds uniform, resulting in an expensive and annoying operating condition requiring a large stock of bearings, pulleys, and bushings for replacement. Upon the decision to reduce the number of diameters of lineshafts used from five to two and to change the motors to drive them approximately at the same speed, the savings made had been sufficient to pay for the change in a year. Mr. Trundle then outlined in detail the steps to be taken in reorganizing a manufacturing organization to reduce delays in delivery and lack of sufficient materials, equipment, and proper tools. By providing gages, reducing the types of product, and standardizing the product, tooling was simplified; standard tools were provided which relieved the pressure on the tool room with a resultant reduction of scrap and also a reduction in working force. He also illustrated how savings in operation might be secured by telling what had been done in the manufacture of automobile jacks. It had been found that jacks of three capacities would be required to handle the full range of loads and a single type of jack had been accordingly designed so that the same patterns and castings could be used for all three capacities the material, however, being varied in each case.

## "Shooting the Wind" to Measure Velocity

A NEW method, simple in principles has been developed for measuring wind velocity and involves the use of a gun. If a gun is pointed vertically upward and a bullet fired from it, the bullet, providing there is no wind, will rise upward, eventually come to rest, and then retrace its path downward until it reaches the muzzle of the gun again. If, however, a wind is blowing, the pressure of the air will deflect the bullet and the latter will fall some distance away from the point from which it was fired. By noticing the distance which such a bullet has been carried away from the starting point of its vertical flight and when knowing certain physical factors, it would be possible to calculate the velocity and direction of the wind, at least up to a certain extent.

Actually, however, instead of shooting vertically and allowing the bullets to fall promiscuously about, the muzzle of the gun is tilted a little toward the wind, just enough to make it possible for the wind to push the bullet back sufficiently to make it fall on or near the steel roof of the shelter from which it was fired. The amount the gun has to be tilted is a measure of the wind's strength.

In order to find the wind velocity at different levels the charge of powder to the gun is steadily increased, the bullet being sent gradually higher and higher.

The results depend a good deal on the selection of the proper size of a bullet and rifle, the present theory applying only to projectiles which leave the muzzle without spin. The greatest height to which the bullets have been fired up to the present is about 720 meters. For this distance a bullet takes about 25 sec. to perform the double journey.—T. Riley in *Discovery*, vol. 5, no. 57, Sept., 1924, pp. 222–223.

# SURVEY OF ENGINEERING PROGRESS

**A Review of Attainment in Mechanical Engineering and Related Fields**

## Symposium on Hardness

AT THE sixth annual meeting of the American Society for Steel Treating, Boston, Mass., September 22 to 26, 1924, an afternoon (Thursday, September 25) was devoted to a symposium on hardness testing. At the time of writing this report only one paper—on The Ball Indentation Hardness Test, by Dr. S. L. Hoyt of the General Electric Co.—had been printed. (*Transactions of the American Society for Steel Treating*, vol. 6, no. 3, Sept., 1924, pp. 396–420, 4 figs., *e.*) Abstracts of the other papers are presented below which were prepared from notes taken at the meeting.

S. C. Spalding, of the Halcomb Steel Co., took as his subject the comparison of Brinell and Rockwell hardnesses of hardened high-speed steel. The tests were carried out on several types of steel with carbon contents varying from slightly in excess of 0.50 per cent to 0.74 per cent, and with tungsten contents varying from less than 16 to more than 18 per cent. The steels also contained chromium and vanadium. A number of curves for the various steels were shown, and from these it would appear that essentially the same general characteristics were exhibited by both the Brinell and Rockwell testers, in the Rockwell case the C scale (the diamond cone) being used. An attempt was also made to establish somewhat roughly the relationship between the values in the two series of tests, a practically straight-line relation being obtained. Attention is called to the fact that the tests described in this case all dealt with hardened high-speed steel exclusively. Among other things, the paper described the methods employed to insure the same hardness conditions in the samples used for the two kinds of tests on each kind of material.

The same subject in its broader aspects was discussed in a paper entitled Relation between Rockwell and Brinell Hardness Scales, by Irving H. Cowdrey, of the Massachusetts Institute of Technology. This paper was based on the work done under the direction of the author by three students at the Institute and covered an extensive series of tests (several hundred in number on a wide-range of materials) ranging from such materials as soft copper and soft lead with a Brinell hardness of 30 up to hard steel and similar material with a Brinell hardness of 800. The application of hardness tests to the extremely soft materials was justified by the desire to establish the relation between the testers rather than to determine the actual hardness of the materials. The tests have been used to establish in the first place the precise relation between the Brinell and the Rockwell scales, and also incidentally the relation between the B (ball) and C (diamond cone) scales of the Rockwell tester itself.

The author does not claim that it has been proved that there is a precise relation between the Brinell and the Rockwell scales, and did not touch at all on the underlying physical foundations that would justify an assumption of the existence of a law connecting the two scales. He states, however, that from these tests the following formulas have been derived (these formulas are purely experimental but have been found to answer the requirements).

The relation between the Rockwell ball scale ($^1/_4$-in. ball) and the Brinell is expressed as follows:

$$R_B = \frac{B - 42}{0.54 - 0.0074B}$$

where $B$ is the Brinell hardness number.

The relation between the Rockwell C scale, or diamond cone, and the Brinell is $R_D = 88.3\,B^{1.61} - 192$.

Among other things, an investigation was made of complaints that the readings on the Rockwell C scale were not always consisting with the results. With this in mind, tests were made on pieces of brass having very coarse crystallization, so coarse in fact that the crystals could be easily seen by the naked eye.

The tests were made by taking from one to four readings within the confines of single crystals and brought out what appears to be a highly interesting situation. In the first place, it was found that while some of the crystals showed a Rockwell hardness of the order of 18 to 20, other crystals gave readings as high as 43, the crystals of such different readings being at times contiguous to each other. This would of course easily explain the lack of internal consistency between Rockwell test readings as compared with Brinell, the difference being due to the fact that the Rockwell C scale indicates the hardness within a comparatively small area tested, while the Brinell with its comparatively large ball gives an average—what might be called an integrated—reading over a fairly large area. The Brinell test is therefore less likely to bring out a possible lack of uniformity in hardness of a material than the Rockwell test.

Another significant fact was brought out in the same series of tests, it having been found that hardness may vary within the confines of the same crystals. In some the hardness at one end of a crystal varied materially from the hardness at the other end; for example, it may be about 31 at one spot and 38 at another, both within the same crystal. The author explains it by offering two hypotheses. One is that a crystal may be twisted and that one reading is taken on the flat of it and the other on the side. Another explanation of the difference of hardness is that in thin materials the hardness reading may be affected by the backing of the material and that, for example, the hardness reading of a metal foil laid on a background of soft copper might be materially different from the reading with the same foil laid over a hardened steel plate.

It would appear from this that the diamond-cone Rockwell hardness tester offers a tool for a possibly fruitful new line of research into the as yet only dimly understood structure of metals.

Samuel L. Hoyt, in his paper on the Ball Indentation Hardness Test, attempts to establish broadly the meaning of the Brinell hardness numbers, basing his work principally on the classical investigation of the Brinell hardness test by Prof. Eugen Meyer.

He starts with an attempt to define hardness. The simplest conception of a hard substance is that it is one which cannot be readily scratched. To go further, however, a distinction must be made between particle hardness and mass hardness. For example, an alundum tube may be easily worn away by a file, but the file, in turn, will be scratched by the particles of alumina which are worn away. The mass hardness of the file is greater than that of the alundum tube, but the particle hardness of the tube is greater than that of the file.

The differences between particle hardness and mass hardness should be considered in connection with the effect of grain size on the hardness. It is inconceivable that a grain of copper, for example, that is a cubic centimeter in size is necessarily any harder or any softer than one a cubic millimeter in size. In massive form, however, from the effect of the bonds between the grains of pure metals, it is quite conceivable that the mass hardness of fine-grained copper differs from that of coarse-grained copper. Again, in alloys composed of two constituents, the constituents may, and generally do, possess different particle hardness. The alloy as a whole has no particle hardness in our sense; instead, each constituent has its own particle hardness which does not vary with the alloy composition, but the mass hardness of the alloy, on the other hand, is a very definite thing—a property that will vary with the composition according to a definite law.

The author comes to the conclusion that for various reasons which he enumerates there is an essential difference between the hardness test and, for example, the tension test. He believes the ball test may be considered as a true hardness test in the light of his definition (not stated explicitly), in particular for plastic and ductile metals.

Meyer showed that the hardness of a metal cannot be truly represented by one figure, as resistance to penetration varies with the degree of penetration of the ball. This variation, however, can be taken into account by the relationship $P = ad^n$, where $P$ is the load in kilograms, $d$ the diameter of the impression, and $a$ and $n$ are constants of the material under test. As the constants $a$ and $n$ belong specifically to a material and the relationship between $P$ and $d$ is not a straight-line one, no one figure can be used to give a true picture of the resistance of the material to penetration, and also no two metals can be compared as to their "resistance to penetration" by means of two figures. Moreover, Meyer has already found that $a$ and $n$ vary quite indiscriminately from metal to metal.

The author discusses next the influence of the ball diameter and points out among other things that the constant $a$ decreases in accordance with an equation given in the original paper with increase in ball diameter for all metals which have an $n$ value that is greater than 2. He shows, however, that the constant $n$ is independent of the ball diameter.

He next derives a law expressing $p_m$ (mean pressure which is supported by the metal, or load divided by the projected area of the impression) in terms of the impression diameter and in terms of the load. If $n = 2$, which is true only for severely cold-worked metals, then the hardness is independent of the load used in the test, and is always equal to $4a/\pi$.

The increase in hardness with penetration is greater the higher the $n$ value, and actually varies from metal to metal.

The present practice of using a constant load of 3000 kg. would be sound only if all metals had equal $n$ values. In general, however, different metals have different $n$ values as well as different $a$ values, and comparisons become more complicated. The author shows how to compare the metals in such cases. The relationships established explain why hardness comparisons made with a load of 500 kg. frequently differ from those which are made at higher pressures, and also why hardness comparisons made on the basis of Brinell numbers are often confusing.

The author proceeds to the discussion of other hardness numbers, referring in particular to the work of Waizenegger and O'Neill. He then proceeds to review the work of Kürth (1908), who brought out some interesting relationships between plastic deformation and the constants $a$ and $n$. Among other things, Kürth showed that annealed metals have a greater capacity for being work hardened than deformed metals, and that numerically this capacity is given by the constant $n$. He also showed that in certain instances it is possible to calculate the yield point of the annealed metal, this being done from the ratio of the change in yield point to the change in hardness. It is pointed out, however, that the yield point, while being definitely related to the hardness, cannot be taken as a measure thereof. As Kürth points out, it is the change in the two properties and not the absolute value of these properties which gives this relation.

Kürth (1909) also examined the effect of temperature on hardness and found it is not true that high temperature by producing an annealing or softening effect would affect the hardening action of the indentation and so show an effect on the exponent $n$. Thus, he found that copper at 500 deg. cent. (932 deg. fahr.), i.e., above the annealing temperature, showed the same hardening on increased loading that was found at room temperature. Various metals do not, however, behave in the same manner in this respect.

The remainder of the paper is devoted to a discussion of the hardness of annealed steel as a function of carbon content, the relation between ball hardness and scleroscope hardness, and the relation between Meyer hardness numbers and Brinell hardness numbers.

The author concludes with a word of caution. Thirty seconds is not long enough to insure getting equilibrium between the load and the sample, and yet one must get equilibrium or these relationships are not obtained. The correct time interval, which may be 10 to 20 min. in some cases, must be ascertained, and will be greater for the softer metals and higher loads. At one time the author tested out this point with a Brinell machine, using a load of 3000 kg., and 30 sec. and 10 min. for the times. The two diameters were 6.63 and 6.73 mm., respectively. This affected the Brinell hardness by only three points, but this error would be greater than is desirable for the determination of $a$ and $n$. On the other hand, the procedure of plotting a number of points on log-log paper gives one a very good insight into the accuracy of his hardness determinations, for the points must fall on a straight line. An inaccurate determination stands out at once. There is no check in present methods. (*Transactions of the American Society for Steel Treating*, vol. 6, no. 3, Sept., 1924, pp. 396–420, 4 figs., *te*)

# Short Abstracts of the Month

## AERONAUTICS (See also Internal-Combustion Engineering)

### Correcting Horsepower Measurements to a Standard Temperature

This report was prepared for publication by the National Advisory Committee for Aeronautics. The relation between the temperature of the air at the entrance to the carburetor and the power developed by the engine is discussed. Its scope is limited to a consideration of the range of temperatures likely to result from changes of season, locality, or altitude, since its primary aim is the finding of a satisfactory basis for correcting power measurements to a standard temperature.

The tests upon which this report is based were made upon aviation engines in the altitude laboratory of the Bureau of Standards. From the results of over 1600 tests it is concluded that if calculations be based on the assumption that the indicated horsepower of an engine varies inversely as the square root of the absolute temperature of the carburetor air, the values obtained will check closely experimental measurements. The extent to which this relationship would be expected from theoretical considerations is discussed and some suggestions are given relative to the use of this relationship in correcting horsepower measurements. (S. W. Sparrow, Mem. A.S.M.E., U. S. Bureau of Standards, Washington, D. C., in Report No. 190 of the *National Advisory Committee for Aeronautics*, 14 pp., 18 figs., 1 table, *e*)

## BUREAU OF STANDARDS (See Engineering Materials; Testing and Measurements)

## ENGINEERING MATERIALS

### Development of Method for Measurement of Internal Stress in Brass Tubing

An investigation recently completed by the Bureau of Mines deals with a new method for the quantitative estimation of longitudinal internal stress in tube shapes, e.g., brass condenser or similar tubing. The various methods that have been applied for the quantitative determination of internal stress in wrought shapes of brass and other copper alloys were found not to be applicable to cold-drawn tubes since the stress in tubes is largely longitudinal. Circumferential stress may be present in tubes, depending upon the mode of manufacture, but longitudinal stress is invariably present. The presence of internal stress in brass tubes can, of course, be demonstrated qualitatively by treatment with an accelerating cracking agent, e.g., a mercury-salt solution or ammonia, but this method is not sensitive to relatively low stresses.

The various methods suggested for stress detection in wrought shapes were applied to leaded brass tubes of the nominal composition 66.33:33.17:0.5 Cu-Zn-Pb, which had been reduced by cold work in the range 17 to 56 per cent. Stress could not be detected by accelerated cracking agents or by the usual cutting methods, but it was known that stress was present because of warping which took place on maturing for long periods at the ordinary temperature. The experiments were carried out on tubes drawn with an internal mandrel.

In the tubes examined it was found that the major stress is longitudinal and that the stress in the outer portion of the tubing is a longitudinal tensile stress, while that in the inner part is a longitudinal compressive stress, the summation of the stresses being zero. Absence of circumferential stress in tubes is shown by the

failure of diametrically cut rings to spring in or out on being slit in two.

The method described for measuring longitudinal internal stress in tubes is called the strip method, and is carried out by slitting a narrow strip longitudinally in a piece of tubing, e.g., a strip 2.75 in. long and 0.10 in. wide in a 3.25-in. tube length, and then releasing one end of such a slit strip by cutting. Stress is indicated by the springing out of the freed end. The stress is calculated by a formula based on the modulus of elasticity of the material and the distance in movement of the freed end, namely,

$$S = \frac{tEi}{l^2}$$

where $S$ = internal stress, i.e., the longitudinal stress
$t$ = thickness of the strip in inches
$E$ = modulus of elasticity of the material
$i$ = amount of springing out in inches, and
$l$ = length of the cut strip.

The slit strip behaves like a beam loaded at one end, and the mathematical development of the formula required for calculating stress is given in the paper.

In the application of the method for determining the effect of low-temperature annealing (without appreciable softening) on stress release, in the case of a tube with longitudinal stress of about 12,000 lb. per sq. in., the stress was reduced about 10,000 lb. by heating for 2.5 hr. at 325 deg. cent. with very small loss in hardness. It has often been thought that because cold-drawn brass tubes do not crack on application of an accelerating cracking agent, they are free from internal stress. The strip method is applicable for the quantitative estimation of stresses inferior to those necessary to cause cracking under the application of mercurous nitrate. This method should be useful in control of manufacturing practice for tubes. (Robert J. Anderson and Everett G. Fahlman. *Abstract of Technologic Papers of the Bureau of Standards, No. 257, et*)

## FOUNDRY

### Die Casting of Aluminum in the Ford Plant

IT is announced that the casting of aluminum in dies has been developed on an extensive scale at the Highland Park plant of the Ford Motor Company. For a long time die casting was looked upon as an impossibility. The old method of casting in sand molds permitted gases to go through the sand as the hot metal was poured in, while pouring metal into dies or solid molds caused air bubbles to form, resulting in so-called "pockets" in the casting, thus leaving an inferior product. Then the secret of feeding the molten metal into the dies from underneath was discovered.

The die is placed directly above the pot containing the hot metal. In fact, it takes the place of a lid. When a casting is to be made all the operator has to do is to turn the air pressure on to the hot metal. The pressure forces the metal up through a feeder into the die, and up to the top of the latter. As the metal goes in, the air is forced out through minute vents provided for the purpose. As the top of the die is filled first, the casting naturally solidifies from that point downward. The air is forced out by the first rush of molten metal to the top of the mold, and as metal only can enter by the feeder, all danger of air bubbles is eliminated and the casting is perfect.

Among the principal parts cast in this department are fan-belt pulleys, the daily production being 10,500. Magnet supports are turned out at the rate of 140,000 each day, while 4500 sedan moldings, 7400 window regulators, and 8000 priming rods are daily averages. (F. J. H. in *The Metal Industry*, vol. 22, no. 9, Sept., 1924, p. 363, *d*)

## FUELS AND FIRING

### A German Research Station for Pulverized Fuel

DATA of experiments of a German research station (locality not named) which now comprises four different pulverizers, two of which work on the "unit" system, delivering pulverized fuel together with a certain amount of air directly into a pipe line and from there into the furnace, while the two other pulverizers deliver by screw conveyors into storage bins whence the fuel is fed into the furnace by a screw feeder and fan.

The superiority of working the furnace from storage bins as compared with the unit system has been proved, because any change in the working of the furnace due, for instance, to the changes of chimney draft reacts on the pulverizer in the case of the unit system and causes fluctuations in the quantity and fineness of the pulverized fuel produced.

The principal problems dealt with concern the size and shape of the combustion chamber and the correct way of introducing the fuel-and-air mixture into the furnace, in order to convey the high temperatures required to the right spot and further to find means for preventing destruction of the firebrick linings of the combustion chamber and furnace.

The author calculates the theoretical amount of what he calls the maximum load of the combustion chamber, which he finds to be practically of the same size irrespective of the kind of fuel used, namely, about 33,5000 B.t.u. per cu. ft. He insists on the combustion chambers' being at least large enough to enable the chimney draft to conduct away freely the gas developed, and that no flames issue out of the ports of the furnace. He demonstrates by a theoretical investigation the necessity of using a long flame, i.e., of burning the fuel gradually as it travels along over the hearth. A drawback of retarded combustion or of using a long flame is that the ash deposits on the hearth. (*The Foundry Trade Journal*, vol. 30, no. 419, Aug. 28, 1924, p. 186, *g*)

### Ash-Softening Temperatures and Clinkering of Coals in a Boiler Furnace

IN TRYING various coals at one of the Government power plants, comparison was made of the ash-softening temperature of each coal with the clinkering tendencies of the coal when burned in the furnace. The proximate analyses, ash-softening temperatures, and plant reports as to clinkering were tabulated. It was desired to bring out any correlation that might appear between the ash-softening temperature as found in the laboratory and the clinkering of the ash as found in the boiler furnace.

Each coal used was sampled and analyzed, and the standard ash-fusion test made. The clinkering of the coal in the furnace was observed by the furnace operator. The furnace equipment consisted of the "V" or Murphy type of stoker with the ordinary firebrick arch roof. With such arrangement very high furnace temperatures are apt to be produced at the ratings carried at this plant.

While the analyses of the different samples of the same coal did not vary appreciably, the ash-softening temperatures were not always so consistent, and the ash-softening temperatures of different samples of the same coal differed as much as 290 deg. fahr. Hence, in order to judge this property of the coal, many samples are desirable, taken from many different shipments.

From tables in the original report there appears a general relationship between the ash-softening temperatures and the clinkering in the furnace. At the top of the table are found the heavy clinkers, and passing downward, conditions improve until no clinkering is found toward the bottom of the table. Too close a relationship could not be expected where this factor alone is taken into consideration. When the ash-softening temperature is properly determined, it will serve well as one of the guides to help in predicting the action of the coal in the furnace. (J. F. Barkley, Mem. A.S.M.E., Fuel Engr., Bureau of Mines, Washington, D. C., in *Reports of Investigations, Department of the Interior, Bureau of Mines*, Serial No. 2630, Aug., 1924, 3 pp., *e*)

## HYDRAULIC ENGINEERING (See also Lubrication)

### The Free Discharge of Large Volumes of Water

EXPERIMENTS conducted by G. R. Collinson, formerly water engineer and manager at Swansea, Wales, with a view to solving the discharge problem in connection with the design of the outlet to the Cray Reservoir of the Swansea Corporation Water Works.

The outlet to the Cray Reservoir is controlled by two 36-in. sluice valves in series, which had never been fully opened since

their installation. On the single occasion when the valves were opened as much as 14 in. a hole was washed in the opposite bank 80 ft. away over the top of a retaining wall 8 ft. high. Subsequently the valves were not opened more than 6 or 7 in. Jet dispersers of the free-vortex type were fitted in turn to the outlet, and their behavior under test is illustrated in figures accompanying the original article. Of these, Fig. 1 shows the hydraulic conditions.

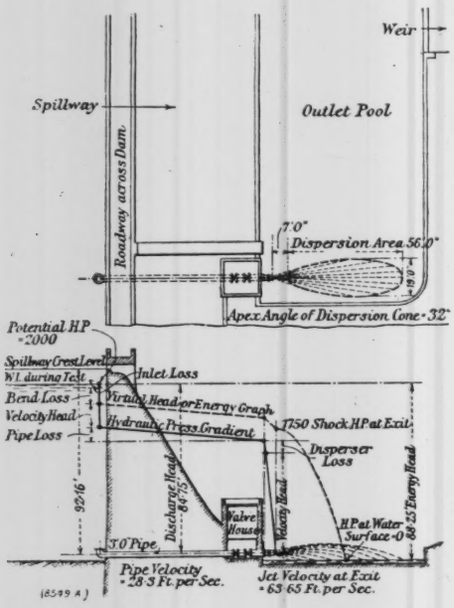


FIG. 1 HYDRAULIC CONDITIONS AT CRAY RESERVOIR, SWANSEA, SHOWING OPERATION OF JET DISPERSERS IN FREE DISCHARGE OF WATER

It was found possible with this apparatus to open the sluice valves to their full extent. One disperser passed about 335 tons of water per min. under a head of nearly 85 ft. This amount of water was dropped over an area of 835 sq. ft. on to the surface of the shallow-outlet pool barely 2 ft. deep, with hardly any disturbance in the pool and without the slightest risk of damage to the surroundings. The water is emitted under these conditions as a solid jet with a velocity of nearly 64 ft. per sec. Before escaping it has imparted to it a free-vortex motion, the center of which rotates more quickly than the periphery. After release the center water takes a more divergent course than the outer water, so that the whole body is a mass of interlacing filaments and is shattered into millions of tiny globules or raindrops. These drops at their high initial velocity are subject to very great air resistance, and they eventually fall harmlessly over an area many times greater than that of a solid jet delivering the same volume.

Taking the average size of the drops as 1/8 in. in diameter, something like 345,000,000 drops would be formed per second. The dispersion area is an approximate ellipse 56 ft. by 19 ft., with a surface of 835 sq. ft. This is 216 times the area of the solid escaping jet, the disperser being 24 in. in diameter. Allowing equal spacing and air gaps, the average number of drops per square foot would be 413,000 per sec., and their average vertical velocity at the water surface rather less than 1 ft. per sec. The apex angle at the disperser is 32 deg. There was no contraction of the escaping jet at the outlet. The diagram shows that 1750 shock horsepower at the outlet were completely dissipated at the water surface.

In Fig. 1 the hydraulic gradient is shown to a scale corresponding to that of the dam. The virtual gradient is to a similar scale but is measured from the downstream water level. The energy present is 22.68 times the ordinate of the virtual head graph.

In the Cray reservoir experiments the disperser proved successful in overcoming the difficult problem of safely releasing large volumes of high-pressure water freely to the atmosphere. Incidentally the very thorough aeration of the previously stagnant impounded water is a valuable feature of the disperser action so far as waterworks practice is concerned. (*Engineering*, vol. 18, no. 3060, Aug. 22, 1924, p. 247, 4 figs., *d*)

### Propeller-Type Turbines of the Hydroelectric Plant at Wynau, Switzerland

DESCRIPTION of a reconstructed plant at Wynau, on the River Aar, where high-speed propeller-type turbines have been installed. The former wheels ran at 42 r.p.m. and delivered 860 hp. The new wheels run at 107 r.p.m. and have an output of 2700 hp. each.

The turbines, the propeller wheels of which are four-bladed, were designed by the Vevey (Switzerland) Machine Co. and are to take care of a head varying from 2.50 to 5.20 m. (8 ft. 2 in. to 17 ft.) and the article describes, and in some parts illustrates, the installation.

The article also reports the data of tests on the first two turbines which were installed in October, 1923, and are claimed to have given excellent satisfaction. These tests (acceptance tests) were carried out in January, 1924, under the direction of the Hydrometric Bureau of the (Swiss) Federal Water Service and the results are shown graphically in Fig. 2 as worked out by the Bureau.

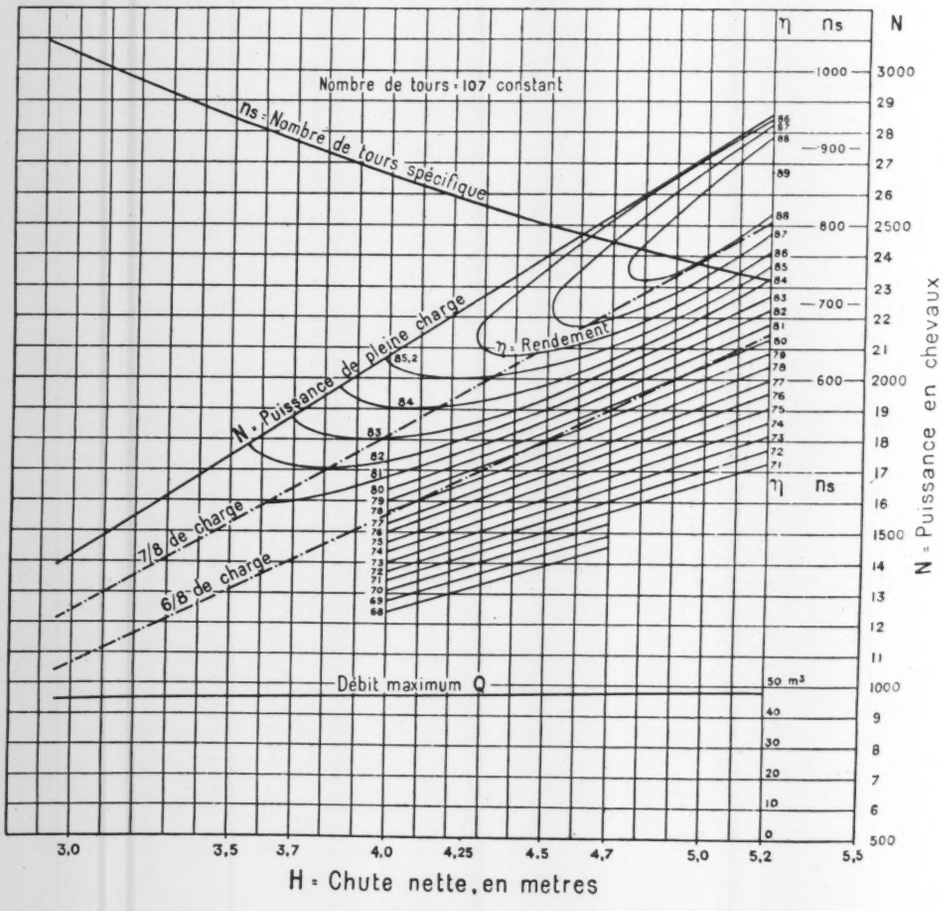


FIG. 2 PRINCIPAL CHARACTERISTICS OF HIGH-SPEED PROPELLER-TYPE TURBINES AT WYNAU

(*Nombre de tours = 107 constant* = revolutions = 107 constant; $n_s$ = *nombre de tours specifique* = specific speed in r.p.m.; *N = puissance de pleine charge* = power at full load; 7/8 *de charge* = 7/8 of full load; 6/8 *de charge* = 6/8 of full load; $\eta$ = *rendement* = $\eta$ = efficiency; *Débit maximum Q* = maximum discharge Q; *N = puissance en chevaux* = N = output in horsepower; *H = chute nette, en metres* = H = net head in meters.)

The maximum efficiency attained under a head of 5.2 m. (17 ft.) is 89 per cent, corresponding to a specific speed of 728 r.p.m. This is all the more remarkable, seeing that the efficiency is as high as 85 per cent when the head has been reduced to 4 m. (13 ft.) i.e., with the specific speed of 862 r.p.m. While the head varies from 2.50 to 5.20 m. (8 ft. 2 in. to 17 ft.) the variations of discharge are only about 8 per cent. This is very important here, because in view of this fact the variations of power output are much less than they would be under similar conditions with Francis turbines. (C. D. in *Le Genie Civil*, vol. 85, no. 7, Aug. 16, 1924, p. 151–154, 8 figs. The same article over the name of R. Hofmann appears as part of a serial in *Bulletin Technique de la Suisse Romande*, vol. 50, no. 17, Aug. 16, 1924, pp. 209–214, 7 figs., *d*)

## INTERNAL-COMBUSTION ENGINEERING (See also Aeronautics; Railroad Engineering)

### New Aircraft Motors

AN ARTICLE dealing with the aircraft-motor situation in general, and with recent developments both for military and for commercial flying.

The purchasers of aircraft motors today are various governments, commercial flying being still in an embryonic stage and taking only a few motors.

A good indication of the state of motor development is given by tests being carried out in the aircraft-motor competition organized in France by the Committee for Aeronautical Propaganda, which latter has offered prizes totaling 2,000,000 francs. Fourteen motors with outputs of 400 to 500 hp. have been entered for the competition and some, having completed the preliminary runs, are now waiting for the 200-hr. continuous test.

It is significant that in none of these motors is there presented any radically new design. All the motors are of conventional types and the performance has to be credited to excellent detail design and a high skill in workmanship and selection of materials.

The author, who was during the war technical expert to the commander of a large aircraft park comprising as high as 25 squadrons, points out an amazing disproportion between the number of planes which were theoretically available in 1918 and the number of those which would actually fly, and further points out that in the last two years of the war the Germans had a far larger percentage of their machines in the air, because of the greater reliability and endurance of their motors. From this point of view, development of commercial aviation is of very great value, as in commercial aviation the planes have to fly all the time and a greater statistical control is kept on the amount of flying and causes of failure.

Commercial aviation requires as a rule that the motors should be capable of flying five, six, or even seven hours (Belgrade to Bucharest, night flight against the wind). Furthermore, commercial aviation requires flights of 300 hr. in three months, and even this is only half what it should be.

It is because of this that when in 1920 and 1921 commercial aviation was attempted, including sections of 500 to 600 km. be covered in one hop, day in and day out, the results were truly disastrous, which can be acknowledged now that the dangers which threatened commercial aviation in those days are no more. It was found, for example, that on certain commercial airplanes which used the same motors with which in 1918 the night bombing planes were equipped, the motors had to be sent to the shops for complete overhauling after every 25 to 30 hr. War reconnaissance planes had motors which lasted in commercial service from 35 to 40 hr., and the author states that it would be more charitable not to say anything about the fragility and unreliability of motors used in pursuit planes.

Since then, thanks to the efforts of commercial aviation, to strict statistics, and to scientific investigations of failure, there has been a complete change in the motor situation, and the motors have been improved at least 100 hp. Rolls-Royce motors will work without overhaul from 140 to 150 hr., while the German motors, though somewhat heavier, are said to be capable of working up to 200 hr. The reliability of the motors has also been greatly improved, and, for example, the aircraft company operating the routes Paris–Warsaw and Paris–Constantinople reports that on flights totaling 1,300,000 km. in 1923 there was only 2.8 per cent of motor failures, and of these a large number were caused by valve failures. There are, however, a much greater number of motor failures on routes requiring long hops—for example, on the Strassburg-Prague route, which, for political reasons, requires a hop of 600 km. Nevertheless, even on this hop, out of 650 flights on French machines there were only twelve forced landings, ten of these being due to motor trouble. Notwithstanding German claims that special new types of motors were used in these planes, it is stated that the regular 400-hp. Lorraine-Diétrich motors were used with only minor improvements. The situation became still more favorable this year, when, up to August, 300 flights all the way across Germany were made without a single forced landing. French commercial-aircraft companies are using exclusively motors dating back to war days. Practically the same situation exists in England and Germany. A number of other motors have been developed, but none have reached the stage where they can be adopted in military aviation.

Commercial aviation in France has adopted extensively the 300-hp. Salmson motor which the army rejected. The majority of new motors are built on the conventional "V" type. However, motors of the "W" and "X" types have appeared, as well as motors with star-disposed cylinders. Cylinders in straight line have also been used where the horsepower does not exceed 300. Efforts to make them more powerful have not been successful.

Among other things, the author recommends coupling "I" motors. In this way, which was one of the methods employed in the French army, designers built up motors successfully up to 700 hp. (Fiat) and even 800 hp. (Sunbeam). A further step is coupling them in "W" and a still further step is the "X" type, such as the Napier "Cub" used by the British Admiralty for day bombers.

The author next refers briefly to four motors of recent design, namely, the 600-hp. Farman, the 450-hp. Lorraine, the three types (260-, 300- and 550-hp.) of Salmson, and the 400-hp. Bristol Jupiter. The Salmson motor has 18 cylinders in "W" and weighs 750 kg. without water, oil, and certain auxiliaries, a comparatively high weight being permitted to make the motor more reliable. The cylinders each have three spark plugs and four valves and the propeller is operated through reduction gears. (Jean Abel Lefranc in *La Nature*, vol. 52, no. 2631, Sept. 6, 1924, pp. 147–157, 10 figs., *dcA*)

### Determination of Gas Distribution in Internal-Combustion Engines by Gas Analysis

IT is claimed that gas analysis enables an engineer to determine the quality of the mixture fed to the different cylinders with a high degree of precision. Briefly, the method consists in making an analysis of the exhaust gas from each cylinder while the engine is in operation.

The present method generally employed to determine distribution is to enrich the mixture fed to the engine until one or more cylinders begin to misfire. This indicates that those cylinders are running "rich." Then the mixture is "leaned" until misfiring again begins, which indicates those that are lean. In a way this method is satisfactory, but unless the ignition is perfect, misleading results will be obtained.

Furthermore, for the determination of distribution a complete analysis of the exhaust is not necessary. All that is required is the percentage of carbon dioxide in the exhaust and the total carbon dioxide in the gas after the sample is burned in a slow-combustion pipette. The method is described in detail in the original paper. To determine the distribution under actual road conditions a 4-cyl. 22.5-hp. engine was tested at two speeds and samples taken simultaneously from the different cylinders. These tests showed that the first (front) cylinder at 15 mi. per hr. received 14 per cent more, the second 9.6 per cent more, the third 12.8 per cent less and the fourth 10.8 per cent less than they should. At 30 mi. per hr. the variation was not so great, yet the loading up of the two front cylinders is indicated, while the two back cylinders received too lean a mixture. The apparatus necessary consists only of a 100-cc. measuring burette, a caustic pipette, and a slow-combustion pipette. It is said that the apparatus is easy to manipulate and dependable results may be obtained by an unskilled technician. (G. W. Jones, W. P. Yant and L. B. Berger, Pittsburgh Experiment Station, in *Report of Investigations, Department of the Interior, Bureau of Mines*, Serial no. 2631, Aug., 1924, 6 pp., 5 figs., *d*)

## LUBRICATION

### Contribution to the Theory of Flow of Viscous Liquids between Journal and Bearings

THE author claims that the equations established for determining the flow of liquids between two eccentric cylinders as worked out by Osborne Reynolds are weak mathematically, because of the numerous factors which he neglected, in particular the leaving out of consideration of members dealing with acceleration. The problem was handled from a more strictly mathematical point of view by A. Sommerfeld and G. Hamel. In the present paper, which is of a thoroughly mathematical character and therefore not suitable for abstracting, the author attempts to present a strictly tenable mathematical solution of the problem, at least in as far as it can be handled as a two-dimensional problem. He does this in part by introducing into the original Reynolds and Sommerfeld formulas a "stream function," and resorting then to the transformation of the equations into isometric coördinates.

Among other things, he finds expression for journal pressures, moment of friction, and the consumption of energy in the liquid film, and discusses in considerable detail the question of negative pressures in the liquid film. This latter is, however, discussed exclusively from a mathematical point of view, the physical meaning of negative pressures in the film of lubricant being practically left out of consideration.

A rather interesting feature of the whole investigation is that in the ultimate conclusion it is found that at least for the case where the differences in the radii of the limiting circles are small, the newly discovered and strictly derived mathematical formulas give results which do not materially differ from the old formulas of Reynolds. This is of value in confirming the fundamental correctness of the older formulas. (Georg Duffing, in *Zeitschrift für angewandte Mathematik und Mechanik*, vol. 4, no. 4, Aug., 1924, pp. 296–314, 10 figs., *tm*)

## MACHINE PARTS (See Testing and Measurements)

## MACHINE TOOLS

### Machine Tools and Their Auxiliaries in the Steel Mills

AN EXTENSIVE article devoted to a consideration of machine tools in general, including historical references, and of their special application to steel mills. The article is really a symposium of several papers, some of which have been prepared by machine-tool manufacturing companies.

In conclusion, figures in the form of tables are submitted representing machine-tool practice in the steel industry. These figures were obtained through a questionnaire sent to about 100 shops. From this it would appear that the lineshaft belt drive is slowly but surely being supplanted by individual motor drive. The figures also show that d.c. is still the most universally used current for operating machine tools. D.c. adjustable-speed motors seem also to be in favor but they do not seem to be making the headway they might, the machine-tool manufacturer still leaning toward the use of gearing to secure the various ranges of speed.

Another table shows that 70 per cent of the motor failures are in connection with motors of over 5 hp. but only 2 per cent of the delays in machine-tool operation are due to motor failures.

To the question as to what is the limiting factor in production, the answers received indicate that the machine tool itself ranks first, the cutting tool second, and the motor third. The general consensus of opinion is that it pays to overmotor machine tools.

As regards standardization the author states: "Very little, if any, attempt has been made to standardize machine tools; neither has there been any effort, at least it does not appear to be in evidence, where the machine-tool industry and the electrical industry have coöperated to any degree which would make it possible for the user of machine tools to interchange control panels or motors.

"I have in mind a large plant which conceived the idea of pulpit control for a large number of machine tools. The pulpits were erected, the framework for the control panels was fabricated and erected. Control panels were purchased for 50 per cent of the shop, but before the job was completed the electrical manufacturers had redesigned the control panels, which made it impossible to use the material which had been fabricated for the balance of the shop.

"This has not only happened in this shop, but a great many instances of this nature could be cited. Sometimes it seems almost a waste of time to try to standardize, in view of the fact that so little coöperation is extended by the manufacturers." (J. F. Kelly, Business Mgr., Association of Iron and Steel Electrical Engineers, Pittsburgh, Pa., in *Iron and Steel Engineer*, vol. 1, no. 9, Sept., 1924, pp. 466–482 and 536–544, 30 pages, *hdA*)

## MARINE ENGINEERING

### Contra-Propellers and Propeller-Guiding Devices

WHILE a great deal has been done in the development of contra-propellers, the actual saving obtained with them has not yet been definitely ascertained. Neither has there been an agreement arrived at as to standardization of guiding devices. The present article is devoted to a description of a number of such devices, particularly to those developed in Germany.

The latest series of trials, executed on a large scale at the Hamburg experimental tank, has proved that the form of the hull also largely influences the propulsive efficiency. Special kinds of propellers and the disk area, pitch, and number of revolutions must be taken into consideration in order to obtain the best results as to working efficiency, saving of fuel, etc., by propeller-guiding devices.

Up to the present time about 130 vessels have been fitted with propeller-guiding devices of the kinds described and, according to American representatives of these inventions, many of them are also in negotiation for American vessels. At present there is being conducted a comprehensive series of model-research experiments, ordered by a prominent American shipping company which is aiming at the best possible results by comparing various devices for their special requirements. These investigations are to be executed in the Hamburg experimental tank, which has a length of 1000 ft., a width of 52 ft., and a water depth of 23 ft. (*Marine Engineering*, vol. 29, no. 9, Sept., 1924, pp. 531–534, 12 figs., *d*)

## MOTOR-CAR ENGINEERING (See also Testing and Measurements)

### New Q.M.C. 4- or 6-Wheel Truck

DESCRIPTION of the designs of Army trucks prepared by the Engineering Section, Motor Transport Division, Quartermaster Corps, U. S. A. These trucks involve a number of new features. Among these may be mentioned rear-wheel brakes which are fully inclosed and operate in oil, an unusual design of winch located so that it is not in the way of any form of body which may be required, seat, dash, and top units which are easily and quickly removable, and a compact form of bevel-gear drive to the front wheels.

Among other unusual features is the use of a universal joint in the steering-gear shaft—a construction designed to insure free steering and facilitate assembly and ready detachment of the steering gear and its column—and a starting motor the function of which is dependent upon an electric-contact device controlled by a sylphon which closes the starting motor circuit only when the engine jacket water is warm. This device is calculated to prevent abuse of the starter and battery by prolonged cranking when the engine is cold. It permits quick cranking when the engine is stalled when warm and also enables cranking under similar conditions if the nose of the vehicle happens to be in a bank or so close to a wall or other obstruction as to prevent use of the head crank.

The band brakes used on the rear wheels are fully enclosed and arranged to run in oil, which makes it possible to employ the brakes for long times on long grades without serious overheating and consequent rapid wear which occurs under similar circumstances with the conventional form of brake. The lining on the heavy brake drum is attached to the internal gear ring instead of being riveted to the brake band.

In the six-wheel-drive jobs both rear axles are substantially identical. They are joined to a drive shaft with two universal joints, by a pair of semi-elliptic springs and by a telescoping torque member of the type employed in the U. S. A. Class C truck shown at Spring Lake in 1923. This torque member is pivoted about vertical

axles at each end and permits the axles to move apart as the springs are flexed, as well as to assume different relative angular positions when the wheels on one side pass successively over uneven places in the road. The tubular torque member is filled with oil which lubricates the turning and sliding surfaces.

The gear set is such that a total of eight forward and two reverse speed changes is available and any of the four speeds of the main gear set can be employed to operate the winch with the truck, either stationary or in motion. This permits employing the winch to pull the truck out of a hole if a suitable anchorage can be found for a rope or cable. (Herbert Chase in *Automotive Industries*, vol. 51, no. 6, Aug. 7, 1924, pp. 271–275, 7 figs., *d*)

## POWER-PLANT ENGINEERING

### Water-Cooled Furnaces at Hell Gate

AMONG the unusual features of the Hell Gate Station is the design of three boiler furnaces having water-cooled side walls (Figs. 3 and 4).

These side walls consist of 4-in. tubes arranged vertically on each side of the furnace and spaced on 7-in. centers. Each tube has two longitudinal steel fins welded on it diametrically opposite each other, and when placed in the boiler furnace the fins overlap,

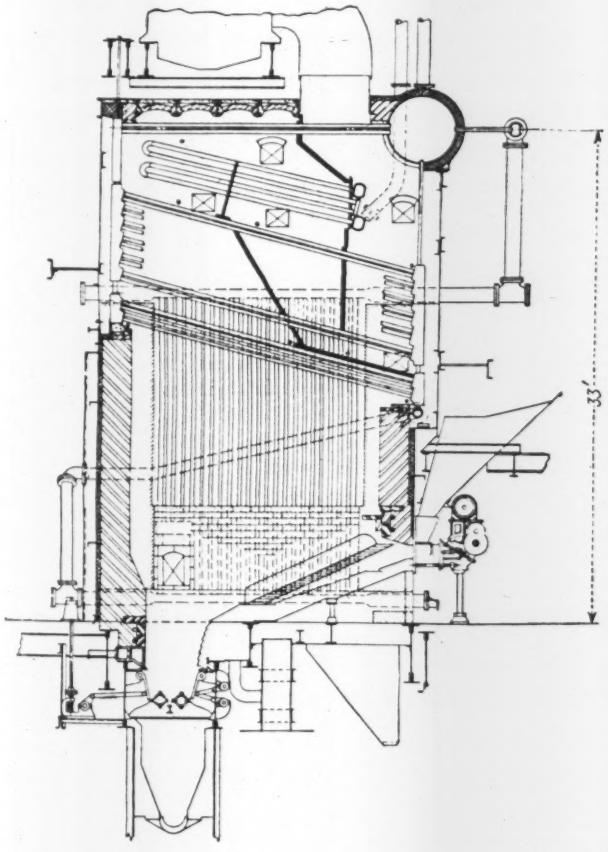


FIG. 3 CROSS-SECTION OF HELL GATE FURNACE WITH WATER-COOLED SIDE WALLS

thus presenting a continuous water-cooled surface to the radiant heat of the furnace. The lower portion of the tube is covered by fireclay tile for a short distance above the stoker. This extends along a horizontal line the entire length of the furnace.

The water in the tubes forms a part of the regular boiler circulation, the lower ends of the tubes being connected by headers leading to the mud drum. The upper ends of the side-wall tubes are connected to a header, which in turn discharges into the boiler drum through a special set of nipples and headers in front of the drum.

The water-cooled side walls not only replace practically all the brickwork in the side walls, but afford considerable protection to the front and rear walls by absorbing the heat which they reflect and thus lowering the temperature of the face of the wall.

One of the boilers equipped with these water-cooled side walls was tested to determine its efficiency and capacity, and the data of tests are shown in the original article.

In one of these tests (Test 5) it would appear that there was a total for controllable losses of only 4.2 per cent. This and the other data indicate that loss of combustible was almost completely eliminated. The very low dry-gas loss was obtained not as much by cutting down excess air as by lowering the flue temperature to 193 deg. fahr. The remarkable feature is that even without the economizer the efficiency was 84.6 per cent, an extremely high figure for a stoker-fired boiler.

The original tests described in a paper by the author of the present article before the Metropolitan Section meeting of the A.S.M.E. in April, 1924 indicate the rating developed by boiler and superheater of 528 per cent. Since these data were presented an 11-hr. test has been made to determine the maximum capacity that could be carried for this period. The average rating was 461 per cent with peaks up to 603 per cent. The limiting factor was the inability

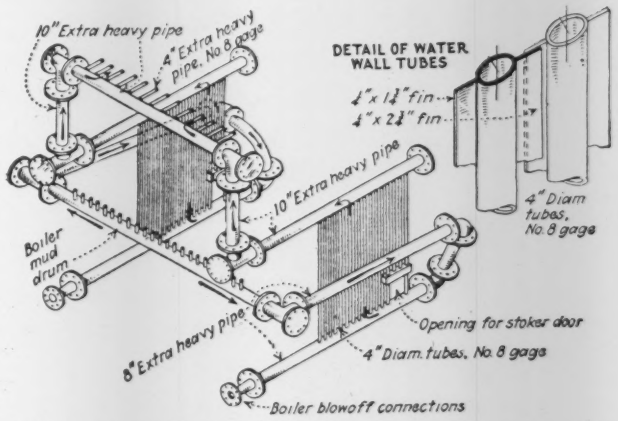
10" Extra heavy pipe
4" Extra heavy pipe, No. 8 gage
DETAIL OF WATER WALL TUBES
4" x 1¾" fin
4" x 2¾" fin
10" Extra heavy pipe
4" Diam tubes, No. 8 gage
Boiler mud drum
8" Extra heavy pipe
Opening for stoker door
4" Diam. tubes, No. 8 gage
Boiler blowoff connections

FIG. 4 ISOMETRIC SKETCHES SHOWING DETAIL OF FINNED TUBES AND METHOD OF CONNECTING INTO BOILER CIRCULATION

to feed more coal and remove the gases with the existing equipment. (W. E. Caldwell, Mem. A.S.M.E., Asst. to the Gen. Supt. of Power Plants, United Electric Light and Power Co., in *Power*, vol. 60, no. 10, Sept. 2, 1924, pp. 354–358, 6 figs., and editorial on page 373, *dA*)

### Water Treatment

DESCRIPTION of exhibits at the Wembley Exposition. Among other things was shown the Chloronome, an apparatus for admitting continuously and automatically a trace of chlorine gas to water, the principal value of which for power plants, especially in hot countries, lies in the ability of chlorine to prevent the formation of deposits of living organisms on the inside of the condenser tubes. These form a slimy and tenacious deposit, which is a strong bar to the conduction of heat. In the chloronome process the amount of chlorine gas does not as a rule exceed one part per 3,000,000 parts of water, even where the cooling water is contaminated by sewage.

A rapid sand filter, called the Paterson, was shown equipped with the so-called "Osilameter" feed gear for the reagents. One of the features of it is an adjustable weir which divides into two parts of any required ratio the flow of the unsoftened water into the plant. The large flow operates the agitating mechanism for keeping the lime cream in constant motion in the storage tanks, and the minor portion is measured by a tip bucket which controls the automatic addition of the lime and soda ash solution in exact proportion to the amount of water flowing, this being done without the use of valves which are always liable to be stopped up by lime. The position of the dividing weir can be altered rapidly by means of a handwheel to give a greater or less dose of reagents according to the quality of the water. (*Chemical News*, vol. 129, no. 3362, Sept. 19, 1924, pp. 174–176, 4 figs., *d*)

### Paper-Tube Thimbles for Pouring Baffles

ONE difficulty which may be experienced with monolithic baffle walls of the vertical type is that of damage to the baffle when

removing and replacing tubes. To overcome this, the Heine Boiler Company now uses a paper thimble, with walls $^1/_{16}$ in. thick, which is slipped over the tube and thus keeps the refractory from touching the tube. When the boiler is fired these tubes burn out and thus leave the tubes free to expand or to be removed without injury to the baffle. These paper thimbles are used with an air-drying cement called "Refractite."

Vertical baffle walls are built up as follows: The paper tubes are slipped over each tube before it is put in place. On each side of the tube bank a wood strip is clamped to the top and bottom tube. This locates the baffle and also serves as the form for the outside edge of the baffle. Boards 1 in. by 6 in. in size, with one edge beveled if they are to be used with a sloping baffle, are passed between the horizontal rows of tubes and held against the side pieces already referred to. Pieces of wood $^1/_4$ in. thick with the ends notched to fit the tubes are placed against the 1-in. by 6-in. boards and between the tubes. This makes a form which is sufficiently right for the use of refractite, which is mixed stiff and tamped into place after each row of tubes is placed. (*Power Plant Engineering*, vol. 28, no. 17, Sept. 1, 1924, p. 925, 1 fig., *d*)

### Operating Code Definitions

Revisions of terms covering both definitions previously adopted by the Prime Movers Committee and terms which have been defined by other organizations. The terms so covered are mainly mechanical, in addition to which, however, the net output of generator is also defined. (Serial Report, Publication no. 24–75, *Prime Movers Committee 1923–24, Technical National Section, National Electric Light Association*, 4 pp., *p*)

## PRODUCER GAS AND GAS PRODUCERS

### The Slagging Producer in Steel Works

Description of the slagging-type producer and data of its operation, together with a discussion of the subject of costs. In reference to the principles of the slagging producer and its early history, compare Mechanical Engineering, vol. 44, p. 600, and vol. 45, p. 657.

There is hardly any dissent today among fuel engineers that the aim in every steel works should be to have coal enter the works at one point only, i.e., the coke ovens, and that the only four fuels should be tar, coke-oven gas, blast-furnace gas, and coke producer gas. The ideal condition would be to utilize tar, coke-oven gas and blast-furnace gas in so economical a manner that it should be possible to discard not only all coal-gas producers, but all auxiliary firing of any kind. In practice such a result can, however, only be obtained where the supply of coke-oven and blast-furnace gas is vastly in excess of the demand and where consequently at least at certain periods profuse bleeding of blast-furnace gas takes place, thus defeating the very economy aimed for.

The difficulty to be contended with is both irregular supply of blast-furnace gas and the greatly varying heat requirements of the works. In fact, at a two-furnace plant the total production of at least one blast furnace must properly be considered as peak, owing to the fact that one of the two furnaces will have longer or shorter periods to be off blast (casting, changing of tuyeres, etc.) and not be producing gas. In such a plant the boilers are really the only place where the gas can be economically used, with the result that at certain times, such as week ends, gas will be bled, and at others auxiliary firing will be necessary.

Such a state of affairs calls for an auxiliary supply of gas to properly equalize the difference between the supply from the furnaces and the demand for heat. Moreover, this auxiliary source of supply should be very flexible, and it is said that the slagging producer answers this requirement.

The principle of the slagging producer is that instead of lowering the temperature in the fire zone, it lowers the fusion point of the ash by adding suitable fluxes and disposes of the ash as liquid slag. The general shape of a slagging producer is that of a small blast furnace, the only difference being that the shaft is cylindrical and considerably smaller. Furthermore, the height of the fuel bed has to be restricted to from 10 to 12 ft., even with the largest type of producer.

On the other hand, it is desirable to have a very generous space on top of the fuel bed for the reception of the large volumes of very hot gas. The area of the hearth is between 0.45 and 0.6 sq. ft. for every ton normal capacity per day. The shaft must be as wide as possible in order to reduce the velocity of the gas, which determines the amount of dust carried out of the producer. For a 200-ton producer the diameter of the shaft is about 12 ft. between linings, and the height of the shaft between 25 and 30 ft. The bosh, which follows the design of the blast furnace, has an angle of 65 to 70 deg. Contrary to the experience with blast furnaces, the lines on which the producers are built are of minor importance.

The whole producer is lined with firebrick, and the lower portion of the shaft shell is water cooled, spray cooling being usually resorted to. As the extent to which spray cooling is effective through firebricks is usually not more than 6 in., the thickness of the lining of the lower portion is limited to this figure.

The hearth is equipped with a number of tuyeres which follow the design of blast-furnace blowpipes. They have to be water cooled. A 200-ton producer will usually be equipped with eight 4-in. tuyeres through which the blast is supplied from a suitable blower. Slide valves or, for preference, automatic check valves between the bustle pipe and the tuyeres prevent the gas from entering the blast pipe. The gas is taken off at the top of the producer through one or more pipes and conducted to a dust catcher on the lines of the customary blast-furnace dust catcher. The charging arrangement follows coal-gas-producer practice for the smaller type of slagging producer, while in the case of the larger types, 200 tons or over, ordinary skip hoists are used. The even distribution of fuel and a uniform height of fuel bed is of minor importance in slagging producers, so that special tops as used for blast furnaces are not necessary.

As regards fuel, it would appear that blast-furnace coke is the most suitable, although where less flexibility is demanded from the producer, other fuels may be used. The smooth and easy operation of the slagging producer is dependent on an easy-flowing and readily melting slag.

As in nearly all fuel ashes the acids (silicates, etc.) greatly exceed the alkalis (lime and manganese), the resultant slag is too sluggish, should one attempt to operate the producer without fluxes. Contrary to blast-furnace experience, limestone or lime is not suitable as a flux. In the first instance, the temperature in the comparatively short gasifying zone, where the reduction between ashes and limestone can only take place, is not high enough to fuse the limestone; secondly, on account of its high contents of CaO, the limestone would form such a small part of the charge that a really intimate mixture with the fuel could not be obtained. Suitable fluxes are blast-furnace and open-hearth slag, which contain the CaO in greatly diluted form and are also more easily reduced.

Where the fuel used is domestic or blast-furnace coke, it is necessary, as will be shown hereafter, to produce considerable quantities of iron in order to make a slagging producer a paying proposition.

The slagging producer is said to produce gas of very uniform composition and heating value quite independently of the rate of operation and the carbon contents of the fuel and the amount of the iron produced. A disadvantage of the gas is its high content of dust, which varies from 1.5 to 4 gr. per cu. ft., depending on the physical qualities of the fuel and flux and the rate of operation. The dust is highly acidiferous (complete analysis given) and contains a very high amount of carbon. One open-hearth furnace was operated for six weeks on uncleaned hot-slagging producer gas and for the first five weeks the results were very satisfactory. Afterward, the use of the gas had to be abandoned because of the checker chambers' being clogged up with the dust. To make it suitable for open-hearth furnaces, the gas will have to be passed through at least a rough cleaning process. So far there is no practical experience available to show that an open-hearth furnace can be operated on cold-cleaned slagging-producer gas. (K. Huessener, Pres. American Heat Economy Bureau, Inc., Pittsburgh, Pa., in *Iron and Steel Engineer*, vol. 1, no. 9, Sept., 1924, pp. 457–464, 3 figs., *dp*)

### New Vacuum Refining Process for Oil

The process consists chiefly in distilling topped crude, the fractions below the gas oil cut, under a high degree of vacuum, 29.9

in., at a comparatively low temperature, say, 630 deg. fahr. It is said that a product is obtained which meets and betters specifications of lubricating oils made by steam distillation, acid and soda treatment, and filtering. The product is said to be used at the plant of the Deepwater Oil Refineries, Inc., Houston, Texas, and at several other plants. In a plant recently installed, the stills instead of resting on the brickwork of the still setting are suspended from overhead structural-steel supports.

The design of the furnace differs from ordinary practice. It is heated by two oil burners, but instead of placing the burners in the usual fashion in the center of the face of the furnace where the flames shoot directly out into the combustion chamber and against the fire sheet of the still, they are placed one on either side of the furnace.

The flames from the burners shoot into tunnels on each side of the furnace. These tunnels, 18 in. by 24 in., run the entire length of the furnace and still, and connect with flue and stack.

They are built of fireproof cement, with perforations along the sides. The flat tops of the tunnels also are perforated with oblong holes just big enough so that each may be covered by a firebrick laid on top of it. As the flame traverses the tunnels, the heat is radiated through the holes into the furnace under the still. This method keeps the flame from striking the fire sheet, which would cause local overheating, cracking of the oil, and early failure of the still bottom.

When the furnace in a new installation is started, all the holes in the tops of the tunnels are closed with firebricks and draft is regulated by removing a sufficient number of bricks to give efficient combustion. The entire furnace is lined with a thick layer of fireproof cement which is pounded into place while in a damp, coarse-powdered form and hardens to the hardness of concrete.

The original article gives details of the working of the process and the properties of the oils obtained. It would appear that the low pressure in the still is one of the most essential elements of the process. (*National Petroleum News*, vol. 16, no. 37, Sept. 10, 1924, pp. 64–66, 1 fig., *d*)

## PUMPS

### Centrifugal-Pump Characteristic Curves

THE author proposes a method of analysis of pump characteristic curves which he considers to be more valuable in actual practice than the present method.

He starts with the critical consideration of two equations, one for the head-quantity curve for a pump at a fixed speed, $H = a + bQ + cQ^2$ ($H$ = head, $Q$ = quantity delivered by the pump, and $a, b, c$ = constants), and the other for the shaft horsepower-quantity curve, $S = d + eQ + fQ^2$ ($S$ = shaft horsepower, $d, e, f$ = constants).

Now, if the above head-quantity curve is correct, it would be natural to expect to find, though of course not necessarily, a term involving $Q^3$ appearing in the power curve. At least its absence ought to be explained. Further, this $H = a + bQ + CQ^2$ curve is suspicious in its simplicity. It should be realized that it is an expression for the "head generated by the pump impeller" multiplied by another expression representing the "hydraulic efficiency." Now, both of these must, of necessity, be complicated expressions owing to the number of factors actually involved, and it would be a matter of surprise if these two complicated expressions, when multiplied together, gave such a simple expression as is usually put forward.

While this type of curve may satisfy an isolated group of experimental readings, the author's experience with hundreds of different pumps is that, in practice, it is too dangerous to use them except with the greatest of care.

The author proceeds to show diagrammatically the conditions at entrance to and outlet from the impeller vanes, and claims that a rational component $w_e$ (Fig. 5) must be taken into account in increasing the head. This rational component $w_e$ is

$$(u_e - v_e \cos \beta_e) \quad \ldots\ldots [1]$$

where $v_e$ = velocity of water immediately after entrance to the impeller and $\beta_i$ = inlet angle of impeller vanes. From this, under certain assumptions, the author derives an expression for the head generated by the impeller. In this expression is used a term $uv \times \cos \beta_e$ where $u$ is the tangential velocity of impeller outlet tip, $v$ the velocity of water before outlet from impeller, and $\beta$ the outlet angle of impeller vanes. The author states, so far as he is aware, that it is always assumed this is settled merely from the dimensions of the impeller, the quantity flowing, and the speed. He therefore lets this term be governed by some factor $c$, the form of which is at present unknown, which somewhat changes his expression for head generated by the impeller. From this he derives the following expression for the shaft horsepower $P$:

$$P = p + \frac{Q}{3300 \times g}\left\{ u^2 - c \times uv \cos \beta - u_e v_e \cos \beta_e + v_e^2 \cos^2 \beta_e \right\} \quad \ldots\ldots [2]$$

where $p$ = horsepower lost in bearings and skin friction, and $u_e$ tangential velocity of impeller inlet tip.

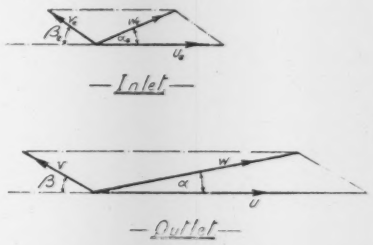


FIG. 5 CONDITIONS AT ENTRANCE TO AND OUTLET FROM IMPELLER VANES IN CENTRIFUGAL PUMPS

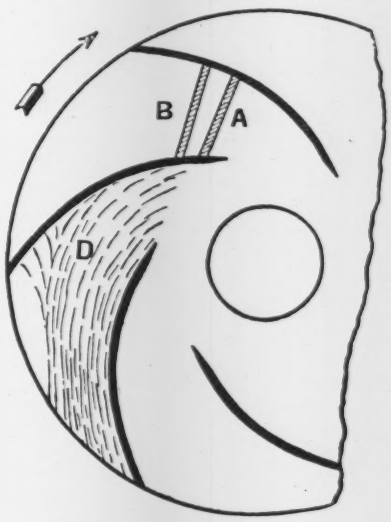


FIG. 6 PUMP DIAGRAM SHOWING FLOW WITH SPECIAL REFERENCE TO FACTOR *c*

Equation [2] suggests a useful method of analysis from the "power-quantity" curves at constant speed. The author considers that these curves are usually more reliable, as a basis of analysis, than the "head-quantity" curves, since it is necessary only to make allowance for bearing and skin-friction losses and leakage losses, both of which can reasonably be assumed to be more or less constant over a wide range of performance of the pump. In dealing with the "head-quantity" curves not only is it necessary to take into account leakage losses, but also the hydraulic efficiency.

Equation [2] involves, as stated previously, a factor $c$, and the author attempts to give it a physical meaning. Referring to Fig. 6, one may imagine the velocity of flow through the impeller to be very low and an elementary block of water $A$ moving very slowly to the position $B$. This being so, the rate at which momentum is being added to the water by the rotating impeller is also very low. In these circumstances it is easy to imagine that the water would flow as a solid mass through the impeller passages and that, in the limit, when the rate of flow is zero, the factor $c$ becomes unity. As the rate of flow through the impeller increases, the rate at which momentum is being added is increased, and one would expect stream lines more as shown at $D$, in which case the factor $c$ becomes greater than unity. Again, if one imagines the rate of flow through the impeller to be so great that the forward moment due to the rotating impeller is just equal to the backward movement due to flow through the impeller passages—in other words, that the absolute path of the water is a radial one and that no momentum is added—in this case the factor $c$ again becomes unity. It would appear, therefore that one should expect a curve somewhat as shown in Fig. 7 when the values of $c$ are plotted against the quantities $Q$ passing through the impeller.

Fig. 8 shows a typical power-quantity curve for a pump running at a constant speed. This is usually a fairly smooth curve until $Q$ approaches comparatively small quantities, when there is a

definite departure from the curve, owing, most probably, to special losses at the impeller tips at small discharge. If, now, this curve be produced backward, the point where it cuts the $P$-axis gives the power $p$ which is absorbed in bearing and skin friction and leakage losses at zero discharge from the pump.

Taking this value of $p$ enables one to determine the value of $(P - p)$ for any quantity $Q$, and hence the factor $c$.

Fig. 9 shows the value of $c$ determined in this way from tests on a 6-stage pump. The cross indicates the position of maximum efficiency point at the speed. This curve should be compared with that of Fig. 7.

More accurate values for the factor $c$ can be obtained if the leakage quantity is estimated at the origin in Fig. 8 shifted back by this amount as done in another figure in the original article. In general, it will be seen that, although if the values of $c$ at small discharges are to be studied, leakage must be taken into account, yet for most practical purposes the rougher method suffices, owing to the fact that the normal range of any pump is on the more or less flat portion of the $c$-$q$ curves. As the factor $c$ has a value of approx imately 2 under normal conditions, it will be readily seen what a great influence a knowledge of its value has upon the design of suitable guide passages, especially as regards their angles. No attempt has been made to do more than indicate a method which, the author believes, is much more valuable than the usual one of employing equations which have proved to be unreliable in actual use. (W. E. W. Millington in *World Power*, vol. 2, no. 8, new series, Aug., 1924, pp. 103–107, 10 figs., *t*)

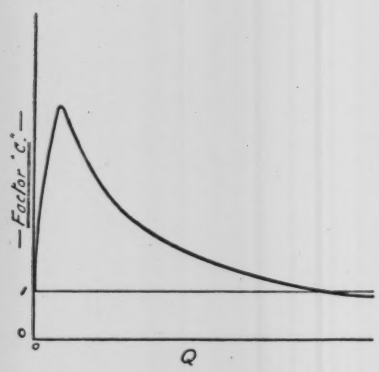


Fig. 7 Values of Factor $c$ Plotted against Quantities $Q$ Passing through Impeller

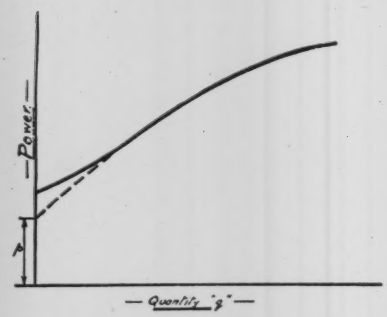


Fig. 8 Typical Power-Quantity Curve for a Pump Running at a Constant Speed

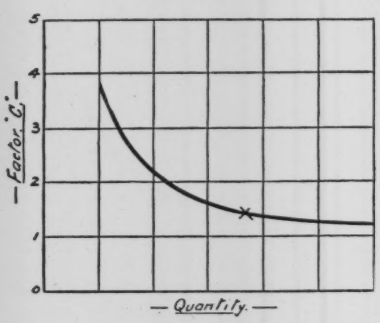


Fig. 9 Values of $c$ from Tests on a 6-Stage Pump

## RAILROAD ENGINEERING (See also Testing and Measurements)

### New Gas-Electric Motor Rail Car

Description of motor rail cars built for the Chicago Great Western Railroad and the Northern Pacific Railway. The power plant consists of a 175-hp. gasoline engine built by the Winton Engine Works in Cleveland, Ohio, and a generator and motors built by the General Electric Company.

In the control of the car no resistances are used and the movement of the controller handle merely connects the motors in series and in parallel for forward or backward movement. The driving of the car is done through the gas-engine throttle, and the electric controller may be left in one position for an entire day's service if no reversals are required. The electric transmission is self-shifting. The windings of the generator are so constructed that the speed ratio between the engine and the car wheels is automatically adjusted to the service the car is called on to perform. The generator is so constructed that the voltage is regulated by the current demands of the motors, and so wound that the product of the voltage and current remains constant in the second power unit.

It is claimed that vibration and noise are absent in this particular car and that the engine and all parts in the engine room are easily accessible. The steel partition between the baggage room and the engine room is bolted to a special frame and can be easily removed. The whole side of the car in the engine room where the compressor is located can be opened up and removed, thus allowing the engine to be taken out of the car for repairing or general overhauling. (*Railway Review*, vol. 75, no. 10, Sept. 6, 1924, pp. 343–348, illustrated, *d*)

### Inside and Outside Drypipes

The inside drypipe is an example of the persistence of an old idea or old form of construction for many years after the reasons for its origin ceased to exist. In the days when a "big" boiler had a shell diameter of 48 in., when there was plenty of clearance for a high dome and very little machinery in the smokebox, it was natural and convenient to pass the drypipe through the top of the shell and the front tube sheets and there attach the steam pipes. If anything happened to it there was nothing to remove at the front end in order to get access to it, but the steam and petticoat pipes, and these, with the niggerhead, merely require the loosening of a few bolts and nuts. The making of drypipe and niggerhead joints was even at that time always a difficult job. Since then, however, the smokebox has been filled with all kinds of devices — netting, diaphragms, valves, pipes, superheater headers, etc. — all in front of and preventing access to the drypipe. Illustrations in the original article show how much dismantling is now necessary in order to do the work on the leaky inside-drypipe joint at the front-tube sheet connection, and the front end is not the only point of attack in this dismantlement, for the inside drypipe must also be freed from its dome connection. In the particular case illustrated the work required to make good a leaky drypipe joint, was equivalent to the time of two men for a period of two weeks.

The advantages of an inside drypipe as compared with an outside, are that it has a natural jacketing of hot steam to protect its contents and is out of sight.

The outside drypipe has been extensively used in Europe and in this country on the Mallet compounds, such as those of the Kansas City Southern. This construction is described in detail in the original article.

Each length of piping is protected by sectional lagging held in place by an outside jacket. The pipe is quite accessible, and, for example, if the dome cracks it can be removed if necessary and repaired without disturbing any section of the drypipe.

The paper concludes by remarking that much is being said in regard to the lack of enginehouse and shop facilities of the railroads throughout the country to properly care for the modern locomotive in its inspection and maintenance requirements. But if more attention were to be given to details of this kind in the designing of new locomotives, it would greatly assist in cutting down the amount of labor required of the roundhouse forces, and result in both a reduction of expense and an improvement in the condition of the locomotives. (*Railway and Locomotive Engineering*, vol. 37, no. 9, Sept., 1924, pp. 261–264, 6 figs., *d*)

## TESTING AND MEASUREMENTS (See also Engineering Materials; Internal-Combustion Engineering)

### Spring-Testing Machine for Railway Rolling Stock

Description of a machine made for the Egyptian State Railways and consisting of a weighing mechanism, upon the table of which the spring to be tested is placed. The pressure is then brought to bear upon the top of the spring through the medium of a double-acting hydraulic cylinder working with oil, the supply being derived from a motor-driven Hele-Shaw pump unit. In using the machine the poise upon the steelyard is set to succeeding increases of load, and the hydraulic ram is brought to bear upon the spring until the steelyard is balanced. The corresponding deflection of the spring for each load increase is then noted. Similar records

are also made for the return stroke of the ram, and the results compared with the former figures. By this relation to the two series of figures the internal friction of the spring can be ascertained.

In addition to the foregoing deflection test, the spring can be subjected to a vibratory test and be compressed solid. This test will at once show if the spring has any tendency to take a permanent set, and also reveal any signs of brittleness due to shock.

For the load-deflection test the pump is operated by a hand-wheel placed in a convenient position for the operator. Sustained loads can be maintained upon the spring by the use of a suitable valve. The deflection of the spring under the applied load is shown upon the graduated scale attached to the main standard of the machine. A dial indicator is also supplied for showing automatically the compression of the spring. Springs in tension can be tested by employing a separate standard which is bolted to the upper surface of the platform, the upper end of the spring being connected to a lug upon the standard, and the the lower end to the crosshead of the ram. (*Railway Gazette*, vol. 41, no. 10, Sept. 5, 1924, p. 313, 1 fig., *d*)

### Tests of Some Girder Hooks

CURVED mild-steel hooks having rectangular cross-sections, designed for 5, 10, and 15 tons, were loaded, using several increments, up to the proportional limit of the material. The stresses on the inside and on the outside surfaces of the hooks were measured experimentally with strain gages having 2 in. gage length. Readings were taken along practically the entire length of the hooks. The stresses were computed theoretically by adding algebraically the component normal to a right section of the hook and the stress due to bending at that section.

The stresses (at the critical section of the hook) were computed by the Winkler-Bach and the Andrews-Pearson formulas. The experimental and theoretical stresses were plotted and compared with the values obtained from the formulas. It was found that the experimental and theoretical stresses agreed very well.

The stresses obtained from the Winkler-Bach and the Andrews-Pearson formulas, which take into account the curvature of the hook, gave correct values for the stress at the critical section. (H. L. Whittemore, Mem. A.S.M.E., Mech. Engr., Bureau of Standards, and Ambrose H. Stang, Associate Physicist, Bureau of Standards, in *Technologic Paper of the Bureau of Standards*, no. 260, 1924, pp. 305–325, 13 figs., *e*)

### Automobile Testing at Mason Laboratory

DESCRIPTION of the special equipment, in particular a rear-wheel dynamometer, installed for automobile testing at the Mason Laboratory, Yale University, New Haven, Conn.

In this apparatus the dynamometer pulleys are suspended from the basement ceiling in such a manner that their tops go through slots in the floor and are approximately level with it. A 15-hp. electric motor is belted to a pulley on the dynamometer shaft and serves as a transmission dynamometer for measuring internal friction of the chassis.

Power delivered by the engine is measured by a 100-hp. friction brake on the drum shaft. The tension of the ropes on the brake pulley is adjusted by a handle at the operator's table and permits a flexible and convenient control of the torque. Fuel consumption is measured by a tank and weighing scale. The scale beam closes an electric circuit which actuates a stop watch and revolution counter, thus recording automatically the time and distance traveled while using up a given weight of fuel. With this fuel-weighing system, runs of only 2 or 3 min. duration have given sufficiently accurate results.

The article outlines a schedule of a rear-wheel test and gives data of a test on a Packard straight-8.

Among other things, the tests have shown that at a speed of 50 miles 33 hp. is required on the level, but the engine horsepower at this speed is over 66, leaving a reserve of 33 hp., or 50 per cent of the total. (E. H. Lockwood, Mem. A.S.M.E., Assoc. Prof. of Mechanical Engineering, Yale University, New Haven, Conn., in *The Yale Alumni Weekly*, vol. 34, no. 2, Sept. 26, 1924, pp. 41–42, 3 figs., *e*)

## STEAM ENGINEERING (See also Thermodynamics)

### A Modified Temperature-Entropy Diagram for a Gaseous Working Substance

THE customary form of the temperature-entropy chart requires, in order to locate any point, a knowledge of the simultaneous ordinate and abscissa values, i.e., corresponding values $T$ and $\phi$. The author made an investigation as to whether the knowledge of pressure and volume per pound of gas would be adequate to determine these values. If the ordinate value is known then the height of the plotted point on the chart is fixed, and it would only remain to fix its abscissa value. If, further, the volume of pressure is known then the abscissa or $\phi$ value can be found by simply logarithmic interpolation from any one of the constant-volume or constant-pressure lines, respectively.

In the equation $PV = RT$, the product $PV$ is proportional to $T$, and hence to the $T$ scale. Generally, pressures are measured in pounds per square inch absolute, and volumes in cubic feet.

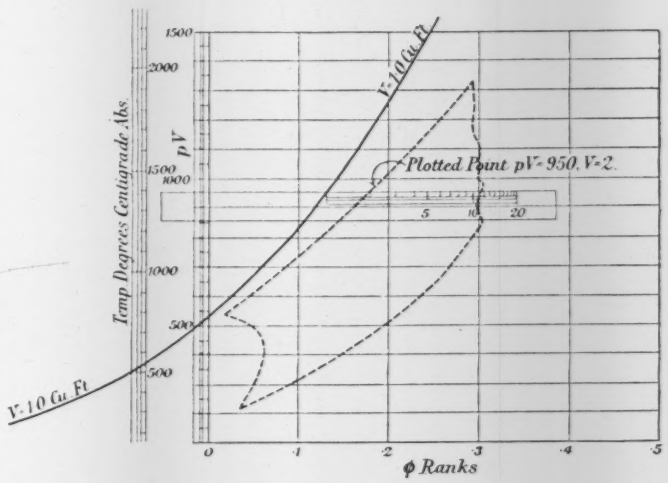


FIG. 10 MODIFIED TEMPERATURE-ENTROPY DIAGRAM FOR A GASEOUS WORKING SUBSTANCE

Then if $R = 99$ ft-lb., the centigrade scale of thermometry being used—

$PV = 99\ T$, where $P$ = pressure in lb. per sq. ft. absolute
or $144\ pV = 99\ T$ where $p$ = pressure in lb. per sq. in. absolute
$V$ = volume per lb. in cu. ft. and
$T$ = absolute temperature, deg. cent.

$$\text{or } pV = \frac{99}{144}\ T$$

$$= \frac{11}{16}\ T$$

From this relation the corresponding scales to be used for the $T$ and $pV$ scales may be determined, and the corresponding value of the $pV$ product for any value of $T$.

If, for example, the $T$ scale is such that 1 in. = 100 deg. cent., then the corresponding $pV$ scale is such that

$$1 \text{ in.} = \frac{11}{16} \times 100 = 68.75\ pV \text{ units}$$

Again, for $T = 400$

$$pV = \frac{11}{16} \times 400 = 275$$

This information enables the $pV$ scale to be constructed and placed correctly relatively to the $T$ scale.

Since the horizontal position of the point may be fixed from a knowledge of the pressure or volume, it is necessary to decide which of these functions should be used. The volumes per pound of gaseous working substances as used in internal-combustion engines lie between 1.0 and 20 cu. ft., and the author uses on the modified $T\phi$ chart a constant line of 1.0 cu. ft. This is the only curve on the chart, and with this is used a simple logarithmic volume scale, shown in Fig. 10. In order to determine the position of a point on

the $T\phi$ chart, it is now necessary to know the $pV$ product and the volume of $V$. The $pV$ product fixes the height of the point, and the $V$ value its position with reference to the 1.0 cu. ft. constant-volume line, using the logarithmic scale to locate the point.

This is followed by detailed instructions for the construction of the chart, such as shown in Fig. 10. The method of transference of indicator diagrams to the chart follows. By this method, described in a paper by Wm. Alexander, An Energy Diagram for Gas Mixture and Some of Its Uses (Proceedings of the Institution of Mechanical Engineers, May, 1917), the corresponding pressures in pounds per square inch absolute and volumes in cubic feet per pound are determined, the volumes being expressed in whole numbers. From this information the product $pV$ is obtained for each point. From the $pV$ and $V$ values the diagram may be plotted with an appreciable saving of time as compared with previous methods.

The author claims for the modified chart a greater simplicity in construction and use, combined with a considerable saving in time. In teaching it has the advantage that each student constructs his own chart, and benefits by the increased knowledge so obtained as compared with the use of a chart which he has not constructed. It may be pointed out that the single constant-volume line used may be done away with by the use of an additional entropy scale on the logarithmic volume scale, but it is not thought desirable to suggest that this should be done.

Experiments carried out to predict the thermal efficiency of an internal-combustion engine from an indicator card by calculating the thermal efficiency given by the transferred indicator diagram on the $T\phi$ chart have proved satisfactory, the agreement between the actual test and the predicted thermal efficiencies being very close. (Prof. G. E. Scholes, Liverpool University, in *Engineering*, vol. 118, no. 3059, Aug. 15, 1924, pp. 215–216, 1 fig., *tA*)

### The Specific Heat of Steam

THE author discusses the variation of specific heat with temperature at saturation and the rational equation for steam up to its critical point. He takes the values of specific heat at high pressures near saturation as an example, because this case has so often been employed in recent years, especially in America, to illustrate in a vivid fashion the difference between different tables and experiments. The differences in the values of the total heats corresponding to the specific heats are, however, relatively small and rapidly become evanescent except near saturation where they are of little or no importance at high pressures.

The main desideratum is a simple and definite standard of comparison, freed as far as possible from minor uncertainties and controversial elements. With this object, the simplest possible form of adiabatic, namely,

$$P(V - b)^{1.3} = \text{constant, or } T/P^{3/13} = \text{constant} \quad [1]$$

has already been adopted by common consent for purposes of reference, and appears to be a remarkably close approximation to the actual facts, though it would be too much to expect that it could be exactly true. According to the equation

$$aP/RT = 1/V - c/(V + b)^2 \quad [2]$$

the value of the index in Equation [1] is still very nearly 1.30 at 350 deg. cent. or 662 deg. fahr. up to 2400 lb., though it sinks rather abruptly to 1.15 at the critical point. Equation [1] requires a *constant* value 0.4772 for the specific heat $S_0$ at zero pressure. According to the experimental data assumed for Equation [2], this is the *mean* value of $S_0$ over the range 212 deg. to 710 deg. fahr. The actual variation of $S_0$ is so small that it can not give rise to any appreciable error in practical calculations provided that the tables employed are exactly consistent with the adiabatic [1].

Next to the adiabatic, the most important relation required for practical calculations is that between $H$ and $V$. In order to be exactly consistent with [1], this relation must be of the form

$$V = (3/13)\,(H - B)/aP + 10b/13 \quad [3]$$

which is very convenient in practice, and extremely accurate to moderate pressures, where it can be verified directly. According to the theoretical equation [2], the approximation is still surprisingly close at the highest pressures near saturation, and gives a value 0.054 cu. ft. per lb. for the critical volume. E. Seiliger (*Zeitschrift des Vereines deutscher Ingenieure*, Jan. 12, 1924) states that "the recent tables of Knoblauch prove the astonishing accuracy of this equation." We should not expect this confirmation to extend to the critical point, where $H$ and $V$ become large and negative.

Some experimental results, given above, have been obtained in an investigation of the specific heat of steam up to the limiting temperature and pressure, now being carried out on behalf of the Electrical Research Association. (Prof. H. L. Callender in the second of a series of articles in *World Power*, vol. 1, no. 6, June, 1924, pp. 325–328, 1 fig., *t*)

### A New Mollier Diagram for Water Vapor

A NEW chart or diagram based essentially on the new steam tables of Knoblauch, Raisch, and Hausen, calculated from specific heats published in Germany in 1923. The present diagram differs, however, in some essential respects from the *JS* total heat-entropy chart published by the three investigators above referred to.

The chart is carried to the pressure and temperature limits of 30 atmos. abs. and 360 deg. cent. (680 deg. fahr.). The author did not follow the lead of Knoblauch and his associates who carried the chart by extrapolation to 60 atmos., and of Stodola who, using the Eichelberg formula, carried his chart to 100 atmos., because, according to his statement, pressures above 30 atmos. are still used only exceptionally for steam-turbine drive, and in the majority of cases he believes that the limits which he uses are sufficiently high for present practice.

He points out further that while for purposes of investigation of the strength and reliability of materials it may be justifiable to make extensive extrapolations, which may also help in investigating the thermal economy of high-pressure steam in turbines, it should not be forgotten that among other things the heat of evaporation $r$ and the specific volume of saturated steam (as recognized among others by Knoblauch) have been experimentally determined only for pressures up to 10 atmos. and temperature of about 180 deg. cent. (356 deg. fahr.). It is known, however, that the value of $r$ materially affects both the location and the course of the upper limit curve, which may be called the backbone of the whole *JS* chart. From this point of view a comparison between the limit curve of the *JS* chart as published by Knoblauch with the same curve in the 5th edition of Stodola's book is instructive. Of similar interest may be a juxtaposition of the heat of evaporation $r$ values in the older steam tables of Schüle and the more modern values of Knoblauch which show, for example, at 25 atmos. values of 447.7 cal. per kg. in one case and 438.3 in the other (805.8 and 788.94 B.t.u. per lb., which means that they differ by no less than 9.4 cal. (16.92 B.t.u. per lb.). As a matter of fact, in so far as heat of evaporation $r$ values are concerned, all the entropy tables that we have are nothing but extrapolations and will continue to be, until the present uncertainty as to the true values of $r$ in the region above 10 atmos. has been eliminated by further experimental work.

As regards the lower limit of pressure, we have to deal in turbine operation with about 0.04 atmos. The $p$ line for this pressure in the region of such a ratio has been carried in the chart down to the initial entropy of 1.46. The $c_p$ values have been experimentally determined only up to 0.5 atmos., and the remainder of the curve has been drawn in the usual manner.

An interesting feature of the present chart is the large scale of ordinates and abscissas on which it has been drawn, it being the belief of the author that such a large scale is needed in a chart intended for use in thermodynamic investigations of steam turbines. (A. Bantlin, Professor at the Technical High School, Stuttgart. Published by Julius Springer, 1924, 2 pp., 1 large chart, *gA*)

## CLASSIFICATION OF ARTICLES

Articles appearing in the Survey are classified as *c* comparative; *d* descriptive; *e* experimental; *g* general; *h* historical; *m* mathematical; *p* practical; *s* statistical; *t* theoretical. Articles of especial merit are rated *A* by the reviewer. Opinions expressed are those of the reviewer, not of the Society.

# Test Code for Speed-Responsive Governors

## Preliminary Draft of a Special Supplementary Code Formulated by the A.S.M.E. Committee on Power Test Codes

*This Test Code for Speed-Responsive Governors has been developed by a Special Committee headed by Francis Hodgkinson as Chairman, the other members being H. Cooke, J. F. M. Patitz, and F. H. Rogers. It is not prepared as an individual code but as an addendum to the codes prepared by the Individual Committees on Steam Turbines, Steam Engines, Internal-Combustion Engines and others having use for speed-responsive regulating apparatus.*

*In 1918 the Committee on Power Test Codes was reorganized by the Council of The American Society of Mechanical Engineers to revise and enlarge the Power Test Codes of the Society published in 1915. The committee consists of a main committee of twenty-five members under the chairmanship of Fred R. Low, and nineteen individual committees of specialists who are drafting test codes for the various prime movers and the other apparatus which constitute power-plant equipment.*

*The Committee and the Society will welcome suggestions for corrections or additions to this draft of its code from those who are specially interested in this subject. These comments should be addressed to the Chairman of the Committee, in care of The American Society of Mechanical Engineers, 29 West 39th Street, New York, N. Y.*

### Scope of Code

1 The scope of this Code is limited to the performance of speed-responsive governors as applied to steam turbines, hydraulic turbines, reciprocating steam engines, and reciprocating oil and gas engines when employed for driving electric generators, mills, etc., or for such uses where the speed-regulation requirements are similar to those for electric generators.

2 This Code takes no cognizance of speed-limiting devices such as safety stops, etc., pressure-responsive pump governors, etc.

3 This Code does not include governor analysis or any discussion of difficulties incidental to speed regulation.

4 Governor tests are carried out for the following purposes:

(*a*) *Speed Regulation.* To determine the speed regulation with variation of load; that is, the ultimate change of speed of the prime mover with one or more changes of load. For convenience the results are commonly expressed in the form of a ratio, as given in a subsequent paragraph.

To determine the speed regulation with constant load; that is, the degree of departure from constant speed at any given constant load with constant operating conditions. For convenience the results are commonly expressed in the form of a ratio, as given in a subsequent paragraph.

(*b*) *Sensitiveness.* To determine the sensitiveness of the complete speed-responsive regulating mechanism; that is, the minimum change in speed at any governor position which will cause a change of governor position.

(*c*) *Stability.* To determine the rapidity with which the complete regulating mechanism controls the prime mover in response to a change of load, i.e., what speed variations occur with sudden application or release of full or partial load, and the time required between the change of load and the settling down of the prime mover to its proper speed with the load to which the unit has been changed. This may also include determination of the rapidity with which the different elements of the complete regulating mechanism are capable of operating.

(*d*) *Governing Power.* To determine the regulating force of the governor and the forces available for operating the valves, or gates, of the prime mover.

### Tests for Speed Regulation

5 *Regulation with Variation of Load.* The regulating characteristics are best determined by operating the prime mover on a load by itself, such as a water rheostat, which may be thrown on and off at will, in whole or in part. The prime mover should be operated with a wide-open throttle, or gate, and under normal conditions as to temperature and pressure at the throttle, and exhaust pressure. The speed to be recorded after various increments of load, commencing at zero and terminating with the maximum load the unit is intended to carry, with not less than eight intermediate increments of load. The load increments should be applied slowly and before taking any readings; ample time should be allowed after changing the load to be sure that the speed is normal for the new steady load and the prime mover has settled down to normal conditions. At each observation the load and speed should be accurately determined. Inlet and outlet water heads in the case of hydraulic turbines and throttle temperatures and pressures and exhaust pressures in the case of steam prime movers should be recorded as evidence that normal operating conditions have been maintained. Other pressures, such as stage pressures within a steam turbine, are illuminative and should be recorded.

6 For this test, speeds may be taken by a mechanical counter and determined by counting the r.p.m. over a period of not less than five minutes. An indicating tachometer should also be used as a check on speed constancy during the period of observation.

7 The above test with ascending load is to be repeated with descending load. The test is to be regarded as unsatisfactory if there is any easily observable hunting of the governor or surging of load while observations are being made, although a slight periodic speed variation not in excess of one-fourth of one per cent in the case of steam prime movers and one-half of one per cent in the case of hydraulic turbines, is not regarded as objectionable.[1]

8 The speed variation between the tested loads may be determined directly from these data. The results of the above tests are to be plotted with load in the abscissa and speed in the ordinate. The curve will show the amount the governor system deviates from the straight-line characteristic.

9 Speed regulation is generally expressed in terms of a ratio and with reference to two extremes of load. Thus the speed regulation between two loads is defined as the ratio of the difference between the extreme speeds and the arithmetical means of the extreme speeds. Thus if

$N_1$ = r.p.m. for load $A$ (the higher speed) and
$N_2$ = r.p.m. for load $B$ (the lower speed) then

$$\text{Speed regulation between loads } A \text{ and } B = \frac{N_1 - N_2}{\frac{N_1 + N_2}{2}} = \frac{N_1 - N_2}{N}$$

where $N$ is the arithmetical mean of the two extreme speeds. It is usual to express this ratio as a percentage.

10 It is seldom, in the case of large electric-generating units particularly, that a unit may be operated alone with means for varying the load. Governor tests must therefore be carried out frequently with the prime mover operating in parallel with other units on commercial loads.

11 The speed regulation test can, however, be carried out if the system on which the prime mover is operating may have its speed or frequently varied by the amount of the probable governor variation. If such a variation cannot be permitted, then all thought of speed-regulation tests must be abandoned. The test may be carried out as follows:

Connect the unit whose performance is to be observed to the power system.

By means of the governor controls of the various units operating bring the frequency of the system sufficiently above normal so that the total variation from normal frequency during the conduct of the test shall be a minimum. This can be accomplished by setting the frequency above normal by one-half the amount by which the frequency of the unit under test may be expected to vary between the two extremes of load under consideration.

[1] Regulation of hydraulic turbines is more difficult than that of steam prime movers because of water's being nearly incompressible, and the action of the governor is affected by eddies, whirls, etc.

While the frequency of the system is being adjusted in this way, set the governor control of the unit under test so that it carries friction load only.

12 When these preparations have been completed, load is slowly shifted in the desired increments to the unit under test by means of the governor controls of the other units which are operating. After the application of each increment of load, the load carried by the unit under test and the corresponding speed should be determined and recorded.

This test should then be conducted in the reverse directions unloading by the desired decrements.

Obviously, the governor adjustment of the unit under test should not be altered in any way during the test.

13 *Regulation with Constant Load.* A test for determining the degree of departure from constant speed of the prime mover with constant load and operating conditions may be approximately determined by employing a sensitive tachometer, belted or otherwise connected to the prime mover, and noting the extremes of speed variations. More accurate determinations may be made with a recording tachometer. For rapid variations, this test had best be made using an electric tachometer generator and oscillograph. The generator should be rigidly connected to the prime mover so that there can be no angular variation between it and the prime mover. Variations of speed under constant load and operating conditions of low period are generally due to defects in the governing mechanism. In the case of hydraulic turbines, small variations of speed of low period frequently occur due to disturbed conditions of flow in the turbine draft tube, particularly at the lighter loads.

Variations of higher frequency also occur in reciprocating engines during a single revolution, caused by uneven turning moments. Such variations and their causes are outside the scope of this Code.

Oscillographs vary greatly in construction, and their descriptions are outside the scope of this Code. The handling of the oscillograph during these tests should be done by one skilled in its use.

The test record shall state:

(*a*) Regulation between....kw. and....kw....., per cent

(*b*) Maximum deviation from constant speed with a constant load of....kw....., per cent.

### Test for Sensitiveness

14 This test is to be made by bringing about a slow change of load and observing the change in speed necessary to cause change of position of the complete regulating mechanism.

This test shall be carried out by graphically recording the change of speed and valve position and by observing the operating conditions, which latter, except the change in load and speed, must be constant.

This test is difficult to carry out with sufficient accuracy to show discrepancies with apparatus that operates well. It serves, however, to show mechanical defects in poor apparatus.

15 A measure of sensitiveness may be made during the speed regulation tests, Par. 3, by noting the difference in speeds between ascending and descending loads. If any observations of sensitiveness are made in this manner, care must be taken that there be no reversal of progress of increasing or decreasing load, i.e., in ascending loads the load must not be first raised and then lowered before readings are taken, and vice versa.

The test record shall state:

Change in speed necessary to cause change of position of the complete regulating mechanism....., per cent.

### Tests for Stability

16 For this purpose a recording tachometer or oscillograph must be employed. The motion of the chart should be approximately one-half to one inch per second. This test will determine the instantaneous variation and the final variation for a given change of load, and the time consumed between the load being changed and the prime mover settling down to constant speed. Prime-mover specifications sometimes state that with certain prescribed instantaneous changes of load the maximum change of speed shall not exceed a specified amount, and that the prime mover will settle down to constant speed within a prescribed time limit. This test gives these data. Sample diagrams illustrating the governor action are shown in Fig. 1, one showing the action in the case of steam prime movers, and another the action in the case of hydraulic turbines.

17 Under this heading are included tests showing the speed with which speed-responsive devices respond to a change in speed of the prime mover and the speed at which relays and the like are capable of operating the valve or gate mechanism throughout their range with the relay operating pressures that are available.

18 One test may be made by means of a multiple-element oscillograph, which can be made to exhibit the time lag of all the involved functions, the respective oscillograph elements being

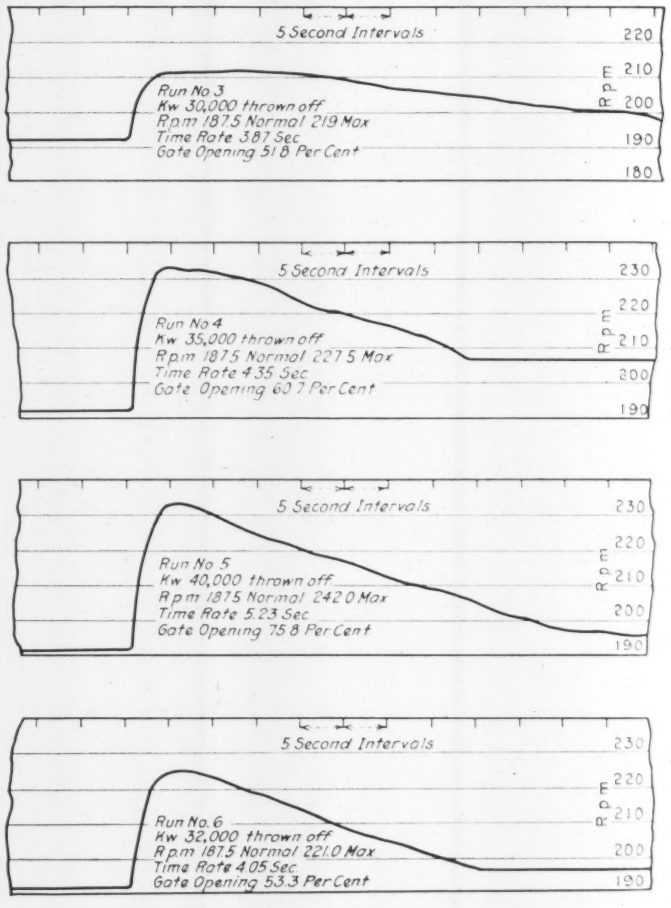


Fig. 1 Diagrams Illustrating Governor Action

arranged to portray the action of such of the following functions as desired and (*a*) and (*b*) being imperative:

(*a*) Time of load change on prime mover
(*b*) Speed of prime mover
(*c*) Movement of control valve or gate
(*d*) Movement of speed-responsive device
(*e*) Fluid pressures operating the relay mechanism
(*f*) Movement of deflecting or by-passing mechanism in the case of hydraulic installations
(*g*) Pressure changes in prime-mover supply in the case of hydraulic installations.

19 The observations of (*a*) and (*b*) are within the scope of this Code. The remainder partake of the nature of governor analysis and are of value, but are beyond the scope of this Code.

20 The oscillograph element indicating load may be standardized to read actual loads. This is not, however, necessary, as the principal function of the oscillograph is to record accurately the time of change. If the element is not standardized, independent load readings shall be made immediately before and after the load change.

21 The report of the test, besides the inclusion of one or other of the above diagrams in Fig. 1, will be a statement that with an instantaneous change of load from (*a*) to (*b*), the maximum speed variation was *N* per cent of the original, and that in....seconds the

## SPEED-RESPONSIVE GOVERNOR TESTS (RESULTS AND DATA)

### GENERAL INFORMATION

(1) Date of test ..... (2) Location .....
(3) Owner ..... (4) Builder .....
(5) Test conducted by ..... (6) Object of test .....

### DESCRIPTION AND DIMENSIONS OF PRIME MOVER AND GOVERNOR

(7) Kind and size of prime mover .....
(8) Type of governor (centrifugal, spring or weight, relay, gear or belt driven, etc.) .....

### TEST DATA AND RESULTS

| | | Beginning of valve opening | No load | Partial loads (Mark normal load with x) | | | | | | Max. load | Extreme valve opening |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | SPEED REGULATION AND SENSITIVENESS | 0 | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | 10 |
| (9) | Column number | | | | | | | | | | | |
| (10) | Load ..... kw. | | | | | | | | | | | |
| (11) | Load—per cent of normal ..... per cent | | | | | | | | | | | |
| (12) | Time ..... min. | | | | | | | | | | | |
| (13) | Atmospheric pressure ..... lb. | | | | | | | | | | | |
| (14) | Pressure at inlet ..... lb. abs. | | | | | | | | | | | |
| | *a*, *b*, *c*, etc. } Pressure at various stages | | | | | | | | | | | |
| (15) | Temperature at inlet ..... deg. fahr. | | | | | | | | | | | |
| | *a*, *b*, *c*, etc. } Temperature at various stages | | | | | | | | | | | |
| (16) | Pressure at discharge ..... lb. abs. | | | | | | | | | | | |
| (17) | Effective head (water turbine) ..... ft. | | | | | | | | | | | |
| (18) | Inlet valve or gate opening ..... per cent | | | | | | | | | | | |
| (19) | R.p.m. increasing load—$N_i$ ..... r.p.m. | | | | | | | | | | | |
| (20) | R.p.m. decreasing load—$N_d$ ..... r.p.m. | | | | | | | | | | | |
| (21) | Mean of 19 and 20 $= \frac{N_i + N_d}{2} = N$ ..... r.p.m. | | | | | | | | | | | |
| (22) | Difference $N_i - N_d = 2\Delta N$ ..... | | | | | | | | | | | |
| (23) | Speed regulation $100 \frac{N_1 - N_2}{\frac{N_1 + N_2}{2}}$ ..... per cent | | | | | | | | | | | |
| (24) | Sensitiveness $\frac{100 \times \Delta N}{N}$ ..... per cent | | | | | | | | | | | |
| | STABILITY | | | | | | | | | | | |
| (25) | Load ..... kw. | | | | | | | | | | | |
| (26) | R.p.m. before load is thrown off ..... r.p.m. | | | | | | | | | | | |
| (27) | Momentary max. speed for sudden decrease of load from no load to part load ..... r.p.m | | | | | | | | | | | |
| (28) | R.p.m. after fluctuations have stopped ..... r.p.m. | | | | | | | | | | | |
| (29) | Time required to reach permanent speed (referring to 27) ..... sec. | | | | | | | | | | | |
| (30) | Maximum increase in speed due to 27 ..... per cent | | | | | | | | | | | |
| (31) | Load ..... kw. | | | | | | | | | | | |
| (32) | R.p.m. before load is thrown on ..... r.p.m. | | | | | | | | | | | |
| (33) | Momentary minimum speed for sudden increase of load from no load to part load ..... r.p.m. | | | | | | | | | | | |
| (34) | R.p.m. after fluctuations have stopped ..... r.p.m. | | | | | | | | | | | |
| (35) | Time required to reach permanent speed (referring to 33) ..... sec. | | | | | | | | | | | |
| (36) | Max. decrease in speed due to 33 ..... per cent | | | | | | | | | | | |
| | RANGE OF SPEED ADJUSTMENT | | | | | | | | | | | |
| (37) | Stroke of clutch or inlet valve ..... in. | | | | | | | | | | | |
| (38) | Clutch or valve stroke, or gate opening ..... per cent | | | | | | | | | | | |
| (39) | Low speed range $N_{min}$ ..... r.p.m. | | | | | | | | | | | |
| (40) | High speed range $N_{max}$ ..... r.p.m. | | | | | | | | | | | |
| (41) | Range of speed adjustment ..... per cent | | | | | | | | | | | |
| | GOVERNING POWER AND WORK CAPACITY | | | | | | | | | | | |
| (42) | Governor force on clutch for . . . per cent change in speed ..... lb. | | | | | | | | | | | |
| (43) | Governor force in low-speed position ..... lb. | | | | | | | | | | | |
| (44) | Governor force in high-speed position ..... lb. | | | | | | | | | | | |
| (45) | Mean work capacity of governor for . . . per cent change in speed or curve of calculated or weighed forces exerted on governor clutch ..... lb. | | | | | | | | | | | |
| | GOVERNOR WITH RELAY | | | | | | | | | | | |
| (46) | Relay fluid pressure ..... lb. per sq. in. | | | | | | | | | | | |
| (47) | Relay piston area ..... sq. in. | | | | | | | | | | | |
| (48) | Force of relay when locked ..... lb. | | | | | | | | | | | |
| (49) | Force of relay available for max. speed motion ..... lb. | | | | | | | | | | | |
| (50) | Time of max. speed motion ..... sec. | | | | | | | | | | | |
| (51) | Work capacity of relay for slow motion ..... in-lb. | | | | | | | | | | | |
| (52) | Work capacity of relay for max.-speed motion ..... in-lb. | | | | | | | | | | | |

prime mover had settled down to a constant speed of $N_1$ per cent of the original.

22 An approximate determination of the speed at which the relay mechanism is capable of operating the control valves may be made with the prime mover at rest and with the relay disconnected from the speed-responsive device by throwing the relay from one extreme of its position to the other and measuring with a stop watch or other timing means the time taken for the mechanism to have completed its travel.

### TESTS OF GOVERNING POWER

23 *Governor Strength for One Per Cent Change in Speed.* This should be determined for the inner and the outer positions of the weights of the speed-responsive device, as in some designs the strength of the governor varies greatly between both of these positions.

24 The governor strength for 1 per cent change of speed may be determined by actual test by operating the prime mover at no load and normal speed and adjusting the linkage between governor and valve gear so that the governor will be close to its high-speed position and the governor control adjusted so the prime mover operates at 99½ per cent standard speed. Then apply a force to the governor clutch and determine the additional force required to bring the speed of the prime mover to 100½ per cent of standard, or 1 per cent higher speed. This can be readily accomplished by measuring the increase of force exerted by the adjustable governor control, as, for example, the auxiliary governor spring. This will be the governor force for 1 per cent change of speed in the high-speed position. This observation to be followed by determining the position of the clutch and then changing the adjustments of linkage between the governor and valve gear until the speed of the prime mover is lowered to 99½ per cent of standard speed. The position of the governor clutch is again recorded and the difference between the two positions corresponds to the governor travel for 1 per cent change in speed in the high-speed position.

25 The linkage between governor and valve gear should next be adjusted so the governor will be close to its low-speed position and the governor control adjusted so that the prime mover operates at 100½ per cent of standard speed. Then, as before, determine the force on the clutch (now in the opposite direction) to bring the speed of the prime mover to 99½ per cent of standard. This, as before, may be conveniently accomplished by measuring the reduction of tension in the control spring. This amount will be the governor force for 1 per cent change of speed in the inner position.

26 With the prime mover operating as above and at 99½ per cent speed, measure the position of the clutch and change the adjustments of the linkage between the governor and valve gear to bring the speed of the prime mover back to 100½ per cent of standard and again measure the position of the clutch. The difference between the two governor positions corresponds to the governor travel for 1 per cent change in speed in the inner position. During these observations, care must be taken that the governor parts at no time come in contact with the governor stops.

27 In carrying out the above test it may be found that for a 1 per cent change of speed the clutch travel will be small and accuracy of measurement not assured. If desired, the speed may be varied in the above manner a greater percentage and the result divided by that percentage to reduce it to a standard of governor strength for 1 per cent change.

28 *Work Capacity for One Per Cent Change in Speed.* This is the product of the arithmetical mean of the two governor forces for one per cent change of speed, as determined under (*A*), and the arithmetical mean of the two clutch travels for one per cent change of speed as determined under (*A*). The test record shall contain the statement:

Governor force for 1 per cent change in speed in the inner position.
Governor force for 1 per cent change in speed in the outer position.
Mean work capacity of governor for 1 per cent change in speed.... in-lb.

29 The governor strength and work capacity may, if more convenient, be calculated by dividing the weights supporting arms and revolving masses into a series of weights and calculating the centrifugal moments of these on the clutch.

30 The integration of the rotating masses as above described is unnecessary in governors where the center of gravity of the weights and arm is approximately in the center of the governor weight. The moment on the governor clutch should be calculated for 99½ per cent speed and 100½ per cent speed, the governor force being the difference between the two.

31 The centrifugal force referred to the clutch may, with advantage, be calculated for a number of positions of the governor travel and plotted on squared paper against the travel of the clutch, when characteristics of the governor may be portrayed.

32 *Strength of Relay Apparatus.* Where relay apparatus is interposed between the governor and control valves or gates, the force that it is capable of giving should be determined by trial. The test should first be by means of a steam-engine indicator, determining the average pressure on both sides of the operating piston while the latter is being caused to travel at maximum speed for the full travel. For this test, with the prime mover at rest, the relay should be disconnected from the governor and the relay plungers quickly thrown from one position to the other. This gives a measure of the force available for motion at maximum speed, this force being the product of the average pressure difference and the area of the piston.

33 The locked force of the relay, that is, the maximum power available for moving the valve gear or gates should they stick, is the difference between the fluid-actuating pressures either side of the operating piston multiplied by the piston area.

34 The test record shall contain the statement:

Force of relay when locked..... lb.
Force of relay available for maximum speed motion..... lb.

35 *The Work Capacity of the Relay* would be the above forces multiplied by the travel of the piston. The test record shall contain the statement:

Work capacity of the relay for slow motion.....in-lb.
Work capacity of the relay for maximum speed of operation..... in-lb.

36 In many cases governors are provided with means by which the speed of the prime mover may be varied some amount above and below normal, which is usually expressed as a percentage. The test may be made by merely observing the extremes of speed which may be attained with the means provided, maintaining the prime mover under the control of the governor. The governor tests should include statements of the following:

Maximum speed at full load for which the governor may be adjusted, ...per cent above normal.
Maximum speed at no load for which governor may be adjusted, ...per cent above normal.
Minimum speed at full load for which governor may be adjusted, ...per cent below normal.
Minimum speed at no load for which governor may be adjusted, ...per cent below normal.

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## The Use of Mercury Vapor for Process Heating Systems

A SUGGESTION has been made that mercury be used in industrial heating systems when high temperatures are needed for process work. Attempts have been made to employ superheated steam, but the objection is found in the fact that the steam loses its temperature very rapidly. Mercury has high boiling points at low pressures. By evaporating mercury in a suitable boiler, the saturated vapor could be led to the heating element and after giving up its latent heat the mercury would return to the boiler. Since mercury shows but a slight temperature change for a large pressure change, temperature regulation would be easily obtained. For most process work the system could be operated sub-atmospheric, thereby eliminating all danger of the mercury's leaking. The problem of boiler construction has been more or less solved in the late developments in the mercury turbine and boiler for power purposes; in fact, the process heating system could be operated in conjunction with a power plant. While a matter for future development, there is reason to believe that such a system will eventually be worked out.—*Power*, Oct. 7, 1924, p. 576.

# Engineering and Industrial Standardization

## Cast-Iron Pipe

ON SEPTEMBER 26, 1924, a meeting was held of representatives of the sponsor bodies for the standardization of cast-iron pipe (the American Water Works Association, the American Gas Association, and the American Society for Testing Materials), to arrive at a plan for representation on the sectional committee, and to get the work under way.

The importance of standardization of materials was one of the matters to which the representatives at the meeting gave special consideration. On account of the lack of definite organization of the producers of cast-iron pipe, the matter of representation of such producers on the sectional committee presents some difficulties; and this matter is one of the topics the conferrees have taken under consideration pending their next meeting.

## Standardization of Tool Holders Begun

A SECOND sub-project under this sponsorship is the standardization of Tool Holders and Tool-Post Openings. To test out the sentiment of the manufacturers toward the standardization of this tool-holding element, the Central Committee on the Standardization of Small Tools and Machine-Tool Elements, Harry E. Harris, Chairman, held an informal conference on the Wednesday afternoon of the New Haven Machine Tool Exhibition. This conference was attended by a considerable number of manufacturers of tool holders, as well as manufacturers and users of machine tools. After a spirited discussion, in which Dr.-Ing. Georg Schlesinger, of Charlottenburg, Berlin, took part, it was unanimously agreed that this sub-project should be undertaken immediately. Accordingly a sub-committee is now being organized so as to be truly representative of the producers, consumers, and general interests.

## Tee-Slot Standardization on the Way

SINCE its organization in April, 1924, the Sub-Committee on Standardization of Tee Slots has been very active. Two regular meetings of its Working Committee have been held, one in Cleveland, Ohio, in May and one in New Haven, Conn., in September. The latter was held at the time of the New Haven Machine Tool Exhibition, which is an annual event at the Mason Laboratory of Mechanical Engineering, Yale University.

The Working Committee has formulated a Progress Report and is distributing it with a questionnaire in an endeavor to bring about complete coöperation on the part of the manufacturers of machine tools using tee slots and to secure additional information which will assist it in framing a final report. This progress report was officially presented to the National Machine Tool Builders' Association at its fall meeting held in Lenox, Mass., October 6–8. This Association is joint sponsor with The American Society of Mechanical Engineers for the Sectional Committee on the Standardization of Small Tools and Machine-Tool Elements, and tee slots is one of the sub-projects being handled by this Committee.

A copy of the progress report and the questionnaire may be obtained upon application to C. B. LePage, Assistant Secretary, A.S.M.E., 29 West 39th Street, New York.

## Standardization and Simplification of Open-End Wrenches

REPRESENTATIVES of the principal manufacturers of open-end wrenches held a conference last month on the simplification of this product. This conference was called by F. D. Mitchell, Secretary-Treasurer of the American Hardware Manufacturers Association, at the suggestion of the U. S. Department of Commerce. Billings & Spencer were represented by Messrs. F. C. Billings, W. S. Swaine and A. W. Fox, J. H. Williams & Co. sent J. Harvey Williams, the Lakeside Forge Co. L. J. Stoddard, and the Vlchek Tool Co. was represented by C. H. Rock. A. E. Foote, of the Division of Simplified Practice, U. S. Department of Commerce, presented the arguments for simplification of variety and size, and offered the services of his Division which is carrying out Secretary Hoover's ideas ont his important subject. F. J. Schlink, Assistant Secretary of the A.E.S.C., was also present, and C. B. LePage represented the Sectional Committee on the Standardization of Bolt, Nut, and Rivet Proportions.

One of the sub-committees of this Sectional Committee is engaged in the standardization of wrench-head bolts and nuts. It has been at work on this problem for something over two years and has endeavored to so select the standard list of "width across flats" that the corresponding number of wrench openings would be reduced. Its present report calls for a 50 per cent (approximate) reduction in the number of wrench openings now in common use.

The conference voted to undertake a simplification program and appointed a sub-committee consisting of Messrs. A. S. Maxwell and W. S. Swaine to follow up the questionnaire which L. J. Stoddard, secretary of the conference, agreed to mail to all manufacturers of forged wrenches.

## A.E.S.C. Developments

IN ORDER to meet the demands made upon it by industry, and to supply the needs of the various working technical standardization committees, the American Engineering Standards Committee has greatly broadened its information services, and has added an engineer translator to its staff for this purpose. In this way, complete information is made available to sustaining members, trade and technical associations and other inquirers on standardization activities in foreign countries, as well as in the United States.

A new development is the appointment of local representatives of the committee in four important industrial centers. These are: K. F. Treschow, Secretary, Engineers' Society of Western Pennsylvania, Pittsburgh; J. B. Babcock, Executive Secretary, Affiliated Societies of Boston; Edgar S. Nethercut, Secretary, Western Society of Engineers, Chicago; Prof. George S. Wilson, Engineering Experiment Station, University of Washington, Seattle.

One of the most striking developments of the standardization movement is the increasingly important role which trade associations are playing in it. More than 140 national trade associations are officially participating in standardization projects under the auspices of the American Engineering Standards Committee. That standardization is a legitimate and constructive activity for associations is everywhere recognized, and explicitly so by a recent decree of the U. S. District Court at Columbus, Ohio.

## Code of Lighting School Buildings

WITH the approval of the Code of Lighting School Buildings as an American Standard by the American Engineering Standards Committee, a demand for definite, detailed, and up-to-date specifications for lighting school buildings, on the part of the architects of school buildings, school superintendents, school boards, and regulatory bodies has been met.

The present code is the result of a thorough revision of the code prepared and issued in 1918 by the Illuminating Engineering Society.

Considerable interest was manifested in the original code, as is evidenced by the fact that it was adopted shortly after its publication by the New York State Department of Education as a guide in planning the artificial lighting of school buildings in that state. The Industrial Commission of Wisconsin used it as a basis for the preparation of the Wisconsin School Lighting Code, effective 1921. Provisions of the code have been incorporated in building codes in several states and municipalities, and have led to marked improvements in school lighting throughout the country.

The new code differs chiefly from the old in being more specific. The illumination standards have been raised to conform with

modern practice; specifications of definite requirements under the glare rule have been included; a limiting ratio of maximum intensity to minimum intensity in classrooms has been included in the rule relating to distribution of artificial light; reflection factors have been specified in the rule relating to color and finish of interior; the rule relating to exit and emergency lighting has been amplified and a rule relating to the illumination of blackboards has been added.

The rules themselves are clear and concise, occupying less than four pages. They are followed by a non-technical discussion of the importance of compliance with them, and directions for carrying them out. These are accompanied by simple diagrams, illustrations of good and bad lighting, etc.

The present code was developed and adopted by unanimous action of a large and representative sectional committee made up of official representatives of the technical, educational, and industrial organizations concerned, acting under the leadership of the Illuminating Engineering Society and the American Institute of Architects.

# Correspondence

*CONTRIBUTIONS to the Correspondence Department of Mechanical Engineering are solicited. Contributions particularly welcomed are discussions of papers published in this journal, brief articles of current interest to mechanical engineers, or comments from members of The American Society of Mechanical Engineers on activities or policies of the Society in Research and Standardization.*

## Proposed Standards for Herringbone Gears

TO THE EDITOR:

The writers of this communication have read with much interest the Report on Proposed Standard for Herringbone Gears for General Commercial Use,[1] and wish to submit certain criticisms.

The report does not define "Herringbone Gear," and definition would appear to be desirable. There seem to be three kinds of herringbone gears manufactured: (*a*) Those built up of two twisted spur gears, the shape of whose teeth are involute in the plane of rotation and the elements of whose teeth are parallel straight lines; (*b*) Those built up of two twisted spur helical gears, the shape of whose teeth are involute in the plane of rotation and the elements of whose teeth are helices; (*c*) Those built up of two helical gears, the shape of whose teeth are involutes when cut by a plane perpendicular to the base helix and the elements of whose teeth are helices.

Type (*a*) is a crude form which has been occasionally used in cast gears and needs little or no consideration in the determination of standards. Type (*c*) results from the use of formed cutters designed for spur gears. Teeth thus shaped do not give proper tooth action. Type (*b*) represents the correct tooth shapes.

It should be pointed out that the enlargements given in Par. 7 of the report are not correct for less than nine teeth. If the involute is laid out and the enlargement added to the addendum, this new circumference will fall outside of the intersection of the two tooth-face involutes in cases of less than nine teeth.

The writers cannot verify the formula for the minimum width of active face given in Par. 10. It seems from mathematical deduction that the value should be about $3.6/P$ instead of $1.6/P$.

In the proposed standard no information is given on the nature of the heat treatment of the chrome-nickel steel for which the allowable stress is given. Likewise further information regarding the condition of the 50 C and 30 C forged steels should be included in the report. If the information given in the report is indeed suitably supplemented from S.A.E. specifications, we obtain the following table for materials and stresses:

| Material | Value of $S$ from report | Yield Point from S.A.E. Spec. Drawn at 1300 deg. fahr. | Yield Point from S.A.E. Spec. Drawn at 800 deg. fahr. | Factor of safety, drawn at 1300 deg. fahr. |
|---|---|---|---|---|
| Chrome-nickel heat-treated steel, S.A.E. 3245 | 30,000 | 98,000 | 200,000 | 3.3 |
| 50 C forged steel | 25,000 | 62,000 | 80,000 | 2.5 |
| 30 C forged steel | 20,000 | 50,000 | 75,000 | 2.5 |

The values for the working stresses given in the table indicate too high a factor of safety. The writers believe that it would be better to state the allowable stress at zero velocity, $S$, in terms of the elastic limit or some other suitable physical property of the material than to give it in specific figures.

No material should be permitted for gears that has no physical properties specified. In the case of the "Cast Steel, A.S.T.M. Class A," for instance, the specifications should be changed to those of "Cast Steel, A.S.T.M. Class B" with an allowable stress $S$ at zero velocity of 20,000 lb. per sq. in.

In designing gears it should be assumed that the gears are designed for their maximum load. On this assumption and basing the factor of safety on the elastic limit of the material, the logical basis, a factor of safety of 1.5 is ample for rolled materials and 2 for cast materials. These factors would materially change the values given for the first three steels in Table 4 in the report.

The writers are not in agreement with the inclusion of the factor $S/2$ in the formula for horsepower rating. It is an acknowledged fact that values such as are recommended above are satisfactory for spur-gear design and that the increase in stress and fatigue with increase in velocity is not greater on herringbone gears than on spur gears. This latter point is taken care of by the constant $K$ used in the report, but if we divide $S$ by 2 we assume that herringbone gears are weaker than spur gears and that the velocity has a greater effect on them than on spur gears. For these reasons the elimination of the division by 2 is recommended.

The values of $Y$, or the shape coefficient, are not correct. First, the values given are based on an addendum of $0.3p$ and dedendum of $0.35p$, where $p$ is the circular pitch. The values of the addendum and dedendum in the report under consideration are $0.255p$ and $0.318p$. Second, the component of the value of $W$ acting normal to a radius and in a plane perpendicular to the tooth is greater than the similar component acting in the direction of rotation, while the tooth shape in the first plane seems weaker than the tooth shape in the plane of rotation. This latter statement has not been proved conclusively, but all investigation seems to indicate such a result.

When Mr. Wilfred Lewis made his original investigation of gear teeth the art of cutting gears was in its infancy, so his report dealt primarily with cast gears. For this reason he assumed that all the load might come on one tooth. At the present time the art of cutting teeth has reached a high degree of development, and it is next to impossible to get all the load to come on one tooth at the point. Considerable work has been done in investigating the effect of more than one pair of teeth in contact at one time, but no concerted attempt has been made to make use of the information obtained. Before adopting a new standard due consideration should be given to this contact coefficient, and it should either be discarded or adopted universally.

In conclusion, the writers would recommend:

1 That the report in its present form be rejected;
2 That the theoretical consideration be more thoroughly analyzed and corrected;
3 That additional experiments be made to determine the effect of velocity on the strength of the teeth; and
4 That additional experiments be made to determine the value of the contact coefficient and that the results obtained be correlated with all previous results.

FRANK A. MICKLE,
CHARLES W. GOOD.

Ann Arbor, Mich.

[1] Prepared by the Sectional Committee on the Standardization of Gears for which the American Gear Manufacturers Association and the American Society of Mechanical Engineers are joint sponsors. In an early form, this report was printed in MECHANICAL ENGINEERING for February, 1923.

## Utilizing the Earth's Heat

[Mr. Trump, whose letter commenting on the article in the August issue of MECHANICAL ENGINEERING dealing with Power from the earth's heat appears below, transmitted with it a copy of his U. S. patent No. 1,258,165 on apparatus for utilizing the heat of liquids and pointed out that this apparatus is similar to the generator suggested by P. Caufourier and described on page 449 of the August issue. The remainder of his comments follow.—EDITOR.]

TO THE EDITOR:

My studies have included the recovery of heat from the earth by circulating water in deep wells located so as to be near enough a complete water supply to give the temperature difference necessary to be efficient.

A study of the entropy diagram shows that the amount of work which can be done by steam between atmospheric pressure and 28 in. vacuum is over one-half of the work which can be obtained from steam of 160 lb. gage pressure to the same vacuum, and a turbine is much more efficient in the lower range than in the upper range.

The circulation of the water after passing through the multiple-effect evaporators into the well would make a very efficient system, especially if the water could be superheated in the earth, and, rising in the central well, will aid in the circulation by forming steam bubbles.

You will note that the system is also capable of furnishing steam for multiple-effect evaporation, which might be useful in the recovery of valuable salts in the water of the hot-well region.

This multiple system recovers 40 per cent more of the heat from hot water than is possible with the single system, and it is possible to use it for the recovery of heat from gases, especially if they contain considerable water vapor, or from a mixture of water and vapor which could not otherwise be utilized.

Syracuse, N. Y. EDWARD N. TRUMP.

# The Thermal Efficiency of Gas-Fired Utensils

IN A THESIS submitted by E. A. Oster, in June, 1924, to the College of Engineering and Commerce of the University of Cincinnati, entitled Observations on the Thermal Efficiency of Gas-Fired Utensils, the author emphasizes the fact that natural-gas conservation is now a question of vital importance in many states of the Union. Owing to its high heating value, its purity and its ease in handling, natural gas is the ideal fuel for home use. Accordingly, with conservation as his incentive, Mr. Oster concentrated his attention on the possibility of devising a thermally more efficient form of standard kitchen utensil.

The thesis presents a wealth of information on the history and literature of this humble but important field of applied thermodynamics.

The experimental apparatus was arranged with great care, and included such refined measuring instruments as an American Junkers calorimeter, thermoelectric couples and auxiliary equipment, platinum foil and gauze targets and an optical pyrometer, all handled with scientific precision. Standard kitchen utensils with ingenious modifications in the bottom of the utensil were used in the investigation.

Some 25 or 26 different forms of utensil bottoms were actually tested. The best of these were probably the forms Nos. 18, 19, 21, and 26, as shown in Fig. 1. The results of the tests are summarized in the following table:

| Design No. | Flame-contact area, per cent of utensil area | Ratio of heat-utilization efficiency to efficiency of plain copper utensil | Heat-utilization efficiency, per cent |
|---|---|---|---|
| 18 | 452 | 2.05 | 67.0 |
| 19 | 552 | 2.09 | 68.4 |
| 21 | 803 | 2.41 | 78.6 |
| 26 | 1063 | 2.42 | 81.8 |

The comparison of these modified bottoms is based on the unaltered copper utensil. The table brings out two facts very clearly: first, that the area available for flame contact must be increased very markedly in order to effect a small increase in the thermal efficiency; secondly, that each added unit of flame-contact area produces a progressively smaller increase in the heat-utilization efficiency. It was further learned in the course of experimentation that the free-flowing radial type of vanes were best as interfering as little as possible with the natural flow of burned gases. The vanes themselves should not exceed $^7/_8$ in. in depth. Their full depth should not extend beyond the diameter of the outer row of holes in the burner used to heat the utensil, and should taper off from this point to a depth of $^3/_8$ in. at the periphery of the utensil.

The data obtained and the conclusions drawn therefrom may be summarized as follows:

1 It is better to use many short flames than a few long ones to heat a utensil.

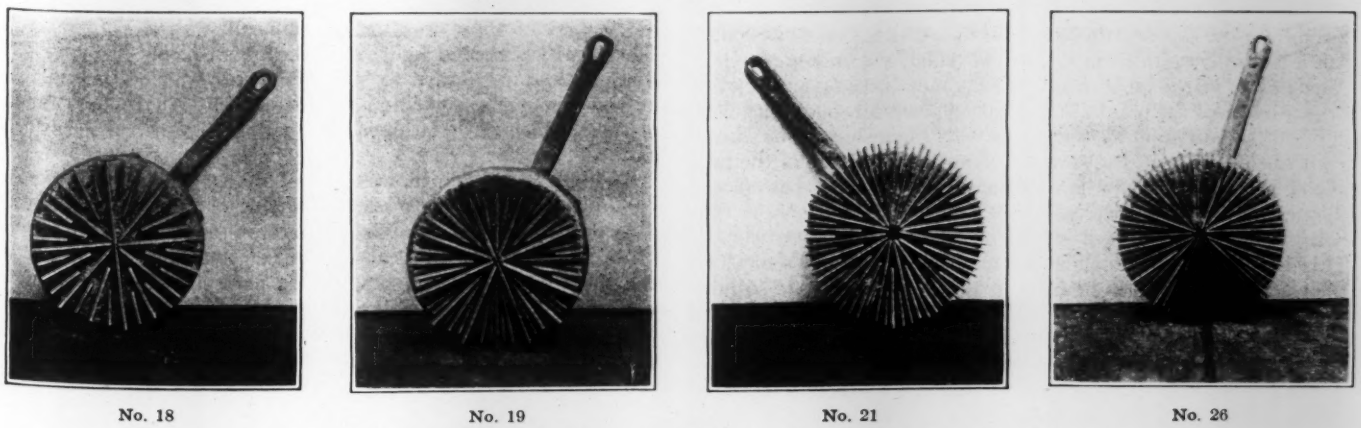
2 The heat-utilization efficiency of a utensil is greatest when the outer blue cone of the flame strikes the bottom of the utensil and spreads out to just the periphery.

3 With existing designs of utensils, the materials composing the utensil has very little effect on the heat-utilization efficiency of the utensil.

4 The heat-utilization efficiency of a utensil increases as the temperature of the heat-receiving surface of the utensil is made to approach the desired temperature of the substance being heated.

5 A utensil can be made with a heat-utilization efficiency 2.4 times that obtained with a standard plain-bottom utensil; that is, equivalent effects can be obtained with a saving of 60 per cent of the gas.

6 A design of heat-receiving surface has been developed which can be applied advantageously to any piece of commercial equipment that is heated by the products of combustion resulting from the burning of any kind of solid, liquid, or gaseous fuel.



No. 18 No. 19 No. 21 No. 26

FIG. 1 DIFFERENT DESIGNS OF KITCHEN-UTENSIL BOTTOMS

# MECHANICAL ENGINEERING

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## Metallurgy for Large Power Plants

THE world has been astounded at the recent rapid developments in power plants. But a few short years ago steam pressures of 100 pounds per square inch and steam temperatures of 330 degrees fahrenheit were considered high. Our modern power plants today are operating at pressures of 400 pounds per square inch and at temperatures of 750 degrees fahrenheit. Regular working pressures of 600 pounds per square inch are being, or will be, used shortly in two or three plants, and there is under consideration the development of a small unit for a working pressure of 1200 pounds per square inch. This unit would have as its antecedent a German experimental plant which operated at 900 pounds and a plant in Sweden which was built for 1500 pounds, though it operated only at 900 pounds.

Picture, if you will, the steam generated in the boiler, then transmitted through suitable fittings to the turbine, from there to the condenser, and then returned through suitable pipe lines to the boiler. Picture the stresses which are set up in the boilers, pipe lines, and turbines due to steam pressure and steam temperature. Realize, too, that when the steam temperature is 750 degrees fahrenheit the metal in the boiler, fittings, and turbine has lost a portion of the strength which it had at atmospheric temperature. Picture further that the metal is held at these temperatures over relatively long periods of time, in some cases running into weeks and months.

Failure to provide suitable material means disaster of the worst order. Not only might it mean a burst boiler or a broken pipe line or turbine; it might mean death for the men in the plant or even the destruction of the plant as a whole. A few years ago, before the tying in of the different systems, this would have meant the throwing out of employment, for the time being at least, of all of the men engaged in gainful occupations dependent upon that power. In fact, one might very properly say that the hands of industry in that locality would have stopped.

This is the picture which every metallurgist should see. This is the responsibility which he faces in advising designers of suitable materials. Yet designers and operating officials are desirous, and quite properly desirous, too, of getting the very greatest efficiency from their plant. Higher efficiencies are obtained through higher operating pressures with resultant higher temperatures. The problem facing the metallurgist, therefore, is to find or develop a material which will be as strong, or practically as strong, at the given temperature as at atmospheric temperature. There is probably no Elysian metal which will meet this condition. There are metals, however, which tend to approach it and those are the metals which should be used. Such metals will probably require heat treatment. This operation should only be performed by persons fully acquainted with these possibilities and limitiations of the operation.

Metallurgically, power plants will probably pass through, in their development, substantially the same periods that the automotive industry has. The automobile demanded exacting requirements. It called for strong metals possessing high ductility. The ordinary metals then available were not suitable. New alloys were developed. Along with these alloys methods of heat treatment were developed. Along with the heat treatment an entire new professional group came to the fore. It is a group we speak of today as heat treaters. Thousands of these men are employed in the automotive industry and hundreds of them possess technical skill and ability of the highest order. No such group has as yet arisen for the power plants, although here and there are isolated individuals who stand out like oases in a desert.

The power plants will require skilled metallurgists or inspectors—call them what you will—in order that they may assure the management that the exacting plant needs have been met. The managers of power-plant-equipment companies will require new corps of trained metallurgists in order to properly produce the equipment needed. Such firms as the General Electric Company, Westinghouse Electric and Manufacturing Company, Allis-Chalmers Company, and others, already have these men. Manufacturers of valves, flanges, fittings, castings, and accessory equipment will require men of this class. Some now have them, although most plants which employ men of this type provide facilities altogether inadequate for proper research. Not one of the plants which it has been my privilege to inspect has provided its men with the same facilities for investigation as the different leading automobile companies have provided for their men.

The metallurgical art for power plants is still backward, although it is true that it has made tremendous strides in recent years. When the day comes when the power-plant industry has technical metallurgists of the same caliber and stamp as are now found in the automotive industry, and when these men have adequate facilities for their work, then we can confidently look forward to real progress in the procurement of suitable materials for power-plant needs.

A. E. WHITE.[1]

## Steel Treating

THE art of steel treating is a very old one. In his Odyssey, in describing the blinding of Polyphemus, Homer says, "And as when a smith dips a great ax or adze in cold water amid loud hissing to temper it—for therefrom comes the strength of the iron—even so did his eye hiss around the stake of olive wood." This shows that even in those dim ages craftsmen knew whence came the strength of steel.

But while the art is a very old one, the industrial applications of steel treating on a large scale are of quite recent date, three developments of the last few years having given it the great and growing importance that it possesses today.

The first of these was the development of the automobile industry, which needed light metal members possessing very great strength and resistance to vibration, and which was controlled by those young enough and progressive enough to dare employ methods which the older men of the industry viewed with suspicion and disfavor.

The next development was that of special furnaces and measuring instruments which made it possible to control heat treating in such a way as to make the product uniform. At first the great objection raised to the heat treating of steel was that it was unreliable, and that while some of the parts thus processed turned out to be of really superior quality, others were weakened rather than improved by heat treatment. However, a better knowledge of heat-treating processes and of their control succeeded in removing this objection.

[1] Professor of Metallurgical Engineering and Director of Department of Engineering Research, University of Michigan. Mem. A.S.M.E.

The third development which facilitated the growth of the heat-treating industry was that of alloy steels, some of which depend for their vital properties upon suitable heat treatment.

The combination of these together with certain other factors which cannot be mentioned here because of restrictions of space, made the heat-treating industry what it is today, and laid foundations for its undoubted healthy progress in the future. The wealth of information in the papers presented at the Annual Meeting of The American Society for Steel Treating in Boston (some of which are abstracted elsewhere in this issue) is but a symptom of this development and an indication of the progress which knowledge regarding metals is making in this country.

## Synthetic Ammonia—A New Task for the Engineer

THE boundary between pure science and engineering or applied science is not a well-defined line but a zone in which the scientist and engineer often work together, and may, in fact, be the same individual. When the social or economic changes of a people or a land demand the removal of some obstacle to growth, the scientist is set to solve the problem. One of the latest of these tasks involves the production of synthetic ammonia, which apparently has passed from the hands of the chemist, the problems of producing it on a large scale being those the engineer must solve.

The United States as it is in its present period of development, has been and is a food-exporting country. Our food exports increased until 1908. Since that time, however, except for a short abnormal period caused by the War, they have declined, and the indications are that if the present rates continue our food exports will cease approximately in 1935, after which the United States will be a food-importing country unless her productivity is increased.

As there are no large amounts of undeveloped land still available, the increase must come through greater yields per acre, which means a greater and more scientific use of fertilizers. It has been publicly stated by an officer of one of our great corporations that it is not impossible that the fertilizer industry may grow to be the greatest industry in the United States.

The three main elements of fertilizer are phosphorus, potash, and nitrogen, the most important of which is nitrogen. The potash and phosphorus situation is still to a large extent in the hands of the chemist. The nitrogen situation, on the other hand, has passed from the hands of the chemist to the engineer. The Claude and Haber processes and their offshoots, the Casale and the Jones processes, are at the present time really engineering problems. By this the writer does not mean that the chemists and the physicists, particularly the latter, are no longer useful in the development of these processes, but that the situation has now reached a point where a large part of the burden is on the engineer and manufacturer. Compressors, circulating pumps, valves, pipes, high-pressure reaction vessels must be developed, and while the American manufacturers and engineers have done well as far as they have gone, there is still an enormous amount of work to be done.

The engineer is on the threshold of a new frontier whose mysteries must be solved.

FREDERICK POPE.[1]

## Patent Office Investigation

SECRETARY Hubert Work recently appointed a committee to study the internal organization and administration of the Bureau of Patents to determine what can be done to simplify and expedite patent procedure and to assure greater accuracy in the work of the bureau. The committee will investigate present methods and hold public hearings for the presentation of suggestions for improvement. Those constituting the committee are Messrs. A. J. Brosseau, U. S. Chamber of Commerce; Jo Bailey Brown, Pittsburgh Patent Law Association; Thomas Ewing, New York Patent Law Association; Harry Frease, Cleveland Patent Law Association; Henry M. Huxley, Chicago Patent Law Association; E. G. Mason, American Bar Association; George A. Prevost, American Patent Law Association; E. J. Prindle, National Association of Manufacturers; Milton Tibbetts, Michigan Patent Law Association; and L. W. Wallace, American Engineering Council.

[1] The Ammonia Corp., New York City.

## Sadi Carnot—The Founder of Modern Thermodynamics—1824–1924

ONE hundred years ago, in 1824, an obscure young French engineer, Nicolas Léonard Sadi Carnot, published a little booklet, which was rapidly forgotten and remained so for a quarter of a century until resuscitated and drawn to the attention of the scientific world by another young but by no means obscure man, William Thomson, the future Lord Kelvin. It was found to contain the fundamental concepts of the theory of thermodynamics and energy which have governed these sciences up to the present day.

It was some time in 1845 that Thomson saw in a paper by Clapeyron a reference to Carnot's work, "Réflexions sur la Puissance Motrice du Feu," but no one could tell him where a copy could be seen. There was none in the library of the Collège de France or in the National Library, and it was not until some years later that he was able to secure a copy of the book, which shows how thoroughly forgotten the work had become.

Lord Kelvin referred later to Carnot's little treatise as an "epoch-making gift to science" and added, "Nothing in the whole range of natural philosophy is more remarkable than the establishment of general laws by such a process of reasoning."

At the time when Carnot wrote there was as yet no clear idea as to the true nature of heat, light, and work, some believing that heat was merely a mode of motion, a form of energy, not only imponderable but actually immaterial, while others thought that heat was a substance which pervaded all bodies in greater or less amount and that heating and cooling were simply the absorption and rejection of this imponderable substance by the body affected. The exact experimental data needed for thermodynamic computations did not appear until a quarter of a century later, as a result of the work of Regnault and his successors. The relation between heat and mechanical energy was only very dimly perceived.

It was under these conditions that Carnot wrote as follows: "One may therefore set it down as a general thesis that motive power is a quantity invariable in nature, and that, properly speaking, it is never either produced or destroyed. True, it may change in form, i.e., it may produce one kind of motion at one time and another kind at another time, but it is never destroyed."

Carnot was the first to declare that "the maximum of work done by heat in any given case of application of the heat energy is determined solely by the range of temperature through which it fell in operation, and is entirely independent of the nature of the working substance chosen as a medium of transfer of energy and the vehicle of the heat."

Because of this Carnot states that "wherever there is a difference of temperature, motive power may be produced;" and further, that "the maximum of power resulting from the employment of steam is also the maximum of motive power that may be realized by any means whatsoever."

He proceeds to a statement that sounds axiomatic today, but was a glance into the far future at the time when it was enunciated: "The motive power of heat is independent of the agents set at work to bring about its realization; its quantity is fixed solely by the temperatures of the bodies, between which in the last analysis occurs the transfer of the caloric. When the gas passes without change of temperature from some predetermined volume and pressure to another equally known pressure, the quantity of caloric absorbed or given up is always the same, no matter what may be the nature of the gas selected as a subject of the experiment."

The importance of the following statement was apparently clearly realized by Carnot himself, as indicated by the fact of its being italicized in his manuscript and book. "The difference between specific heat under constant pressure and specific heat under constant volume is the same for all gases." He proves it essentially in the following manner. The increases of volumes with the same differences of temperature are the same according to Gay-Lussac and Dalton, and therefore, according to the laws of thermodynamics

as demonstrated by Carnot, the heat absorbed with equal augmentations of volume being the same, the two specific heats are constant and their difference as well. Carnot goes even further and bases upon this principle a determination of the specific heats at constant volume, taking as his values of the determined specific heats of constant pressure those of Delaroche and Bérard; these make the constant difference 0.300 with air at constant pressure being taken as unity.

Carnot realized the relation of heat energy to work done and states: "The caloric head produces more motive power in the lower degrees of temperature than in the higher degrees of temperature," a fact which is clearly understood today as being due to the circumstances that one degree at low temperature and the corresponding quantity of heat are larger fractions of the total temperature and the total heat stored in the substance than one degree at a higher point on the scale of absolute temperature.

Two other quotations may be made from Carnot's work (quoted through the translation made by R. H. Thurston, 2nd edition, John Wiley & Sons, New York, 1897).

When an elastic fluid passes without change of temperature from the volume $U$ to the volume $V$, and when a similar ponderable quantity of the same gas passes at the same temperature from the volume $U'$ to the volume $V'$, if the ratio of $U'$ to $V'$ is found to be the same as the ratio of $U$ to $V$, the quantities of heat absorbed or disengaged in the two cases will be equal. This theorem might also be expressed as follows: When a gas varies in volume without change of temperature, the quantities of heat absorbed or liberated by this gas are in arithmetical progression, if the increments or the decrements of volume are found to be in geometrical progression.

The other quotation will be of interest to all those engineers who are today considering the employment of higher and always higher steam pressures, and who are multiplying the stages of steam turbines.

It is easy to see the advantages possessed by high-pressure machines over those of lower pressure. This superiority lies essentially in the power of utilizing a greater fall of caloric. The steam produced under a higher pressure is found also at a higher temperature, and as, further, the temperature of condensation remains always about the same, it is evident that the fall of caloric is more considerable. But to obtain from high-pressure engines really advantageous results, it is necessary that the fall of caloric should be most profitably utilized. It is not enough that the steam be produced at a high temperature; it is also necessary that by the expansion of its volume its temperature should become sufficiently low. A good steam engine, therefore, should not only employ steam under heavy pressures, but under successive and very variable pressures, differing greatly from one another, and progressively decreasing.

The originator of what is known today as the Carnot cycle was born June 1, 1796, in the smaller Luxembourg palace, Paris, while his father was a member of the Directory then holding the executive power in France. Curiosity, especially in regard to mechanics and physics, was one of the essential traits of his mind, and on account of this disposition the elder Carnot did not hesitate to give a scientific direction to the boy's studies. He was educated in the Polytechnic School, which he entered in 1812 at the age of sixteen. This institution at that time was preparatory to military service. Later he occupied several comparatively unimportant positions in the French engineering service, largely due to the fact that neither the Napoleonic government nor the Bourbons who succeeded it, viewed the family antecedents of the young man with any too great favor. Carnot died from an attack of cholera on August 24, 1832.

The following anecdote may give an insight into the personal character of the man. It is true that it refers to that period when Carnot was only about four years old, but it should be remembered that he died a comparatively young man.

While Napoleon Bonaparte was First Consul, Carnot's father was his Minister of War, and as such had a free entry to the Malmaison Palace and indeed became a favorite of Madame Bonaparte, the First Consul's wife. The latter was one day with some other ladies in a small boat on a pond, the ladies rowing the boat themselves, when Bonaparte, unexpectedly appearing, amused himself by picking up stones and throwing them near the boat, spattering water on the fresh toilets of the rowers. The ladies dared not manifest their displeasure, but the little Sadi, after having looked on at the affair for some time, suddenly placed himself boldly before the conquerer of Marengo, and threatening him with his fist, cried, "Beast of a First Consul, will you stop tormenting those ladies!"

## The A.S.M.E. Annual Meeting

THE October 22 issue of the *A.S.M.E. News* contained the tentative program for the coming Annual Meeting of The American Society of Mechanical Engineers, which will be held in New York, December 1 through 4. An important innovation in connection with this meeting is the manner in which printed copies of the papers will be distributed previous to the meeting. In this issue of MECHANICAL ENGINEERING, on pages 726–729, appear synopses of papers that will be printed in advance in pamphlet form. A suitable blank on which to order these papers will be found on page 104 of the advertising section. The remainder of the papers will appear in a special number of MECHANICAL ENGINEERING which will be mailed to the membership on November 15 in ample time to reach the farthest point from New York in the United States at least a week before the meeting. The October 22 issue of the *News* also contains a blank for ordering papers that are printed in pamphlet form. Those members who do not desire to bring their copies of the special issue of MECHANICAL ENGINEERING to the meeting with them, may procure additional copies which will be on sale at headquarters during the meeting.

The technical program for the coming meeting contains a well-balanced list of strong papers. There will be the usual representative papers of interest and value to the power-plant field, which will be presented in four sessions: one on oil burning, one on steam power generation, one on oil engines and gas turbines, and one on hydraulic-power-plant problems. There will also be an impressive array of titles of interest to the engineer who is interested in machine design, machine-shop practice, and management, including those of papers presenting new information about mechanical springs, lubrication, methods of measuring hardness of metals, and gears. Members of the Society interested in management will enjoy the session at which Taylor's classic paper on shop management will be re-presented, and by papers dealing with production control, the design and manufacture of a standard machine, and management development in a modern hosiery plant.

Strength and Proportions of Wheels, Wheel Centers and Hubs, Discharge of Flow-Measuring Nozzles for Air, and The Temperature at which a Liquid Evaporates into a Gas, are subjects of papers which will make up another session of general engineering interest. Three valuable papers on Research and Lubrication will furnish information of importance to designing engineers. The National Defense Division will open its session with an address by Assistant Secretary of War Davis on the Engineering of National Defense. The remainder of the session will be taken up with a consideration of technical problems of ordnance design and ordnance materials.

The Railroad Session will be devoted to a discussion of the Zoelly turbine-driven locomotive. The Textile Division will discuss the development of the spinning frame and the engineer's field in industrial economics. Joint sessions with the American Society of Refrigerating Engineers and the American Society of Safety Engineers will be held during the meeting.

At the business meeting the presentation of the bust of Admiral Melville, Past-President of the Society and pioneer in the development of steam-driven naval vessels, will be made.

A novelty in the technical program is the addition of two lectures, one bordering on the field of economics, and the other on the field of pure science. At the economics lecture Dr. Julian D. Sears will discuss the petroleum situation in the United States.

A number of interesting innovations will feature the program of the social events at the meeting. The importance of this gathering as an opportunity for developing professional fellowships and cementing friendships will be emphasized. As now planned, the social events will consist of Open House on Monday evening, December 1; the Presidential Reception following the Presidential address on Tuesday evening; the Ladies' Tea and Reception on Wednesday afternoon; and the Annual Dinner on Wednesday evening. Elaborate arrangements are being made to entertain the ladies who accompany their husbands to the meeting. The principal innovation is the Annual Dinner on Wednesday evening, to which the Committee on Meetings and Program is attaching great importance as a possible climax of Society enthusiasm. There will be one speaker of international reputation, after whose address

the evening will be devoted to greeting the men who have joined the Society during the preceding year and who have not had the opportunity of enjoying the Annual Meetings of the Society.

The Third Annual Exposition of Power and Mechanical Engineering will be held in the Grand Central Palace, New York City, from December 1 through 6, parallel to the meeting. During the past two years this exhibition has increased in scope and interest until it embraces all types of equipment in the power mechanical-engineering field. The coming event will include the showing of nearly three hundred manufacturers' exhibits and will occupy 150,000 sq. ft. on three floors of the Grand Central Palace. Among the types of apparatus that will be displayed are included steam boilers, superheaters and economizers, air preheaters, stokers, prime movers, pipes, valves and fittings, ash- and coal-handling equipment, flow- and temperature-measuring devices, mechanical power-transmission equipment, machine tools, heating and ventilating equipment, and refrigerating machinery. The value of the coming exposition will be increased by the addition of a series of lectures on important phases of the development in the mechanical-engineering field. Leaders in this development will talk on the boiler room, steam prime movers, oil and gas engines, hydroelectric power-plant equipment, materials handling, modern machine-tool developments, mechanical power transmission, mechanical refrigeration, heating and ventilating. These lectures will be arranged so that they will not conflict with the more formal papers being presented at the Annual Meetings of the A.S.M.E. and the A.S.R.E. There will also be a series of interesting technical moving pictures.

Headquarters will be established for the members of the A.S.M.E. in a booth handy to the entrance of the Power Show, where stenographic service and information about the exposition and the Society meeting will be available.

## Distinguished European Engineers Entertained

THE Presidents and members of the governing bodies of the Four Founder Engineering Societies entertained Sir Charles Algernon Parsons and Senator Luigi Luiggi at a luncheon at the Engineers' Club, New York, on Friday, September 26.

About eighty leaders of the engineering profession were present to greet the distinguished engineers. John W. Lieb acted as toastmaster and introduced representatives of the four societies, who gave warm words of welcome. Past-President Loweth represented the A.S.C.E., Vice-President J. V. W. Reinders, the A.I.M.E., President F. R. Low the A.S.M.E., and President Farley Osgood the A.I.E.E. The speakers emphasized the cordiality with which the American engineers had been greeted in Europe the past summer and spoke of the importance of international contacts between engineers of all countries. Telegrams of regret were read from Sir Charles L. Morgan, president of the Institute of Civil Engineers of Great Britain, and William H. Patchell, president of the Institution of Mechanical Engineers of Great Britain, who were unable to be present.

After responding to the words of welcome, Sir Charles A. Parsons paid a tribute to the tremendous strides in power development that were being achieved in this country and forecast even greater developments of steam-produced power to meet the growing demands of our civilization. He then spoke of the coöperation of engineers and scientists in a project to investigate the depths of the earth by sinking a well twelve miles deep. He pointed out that such an investigation would be of more practical value than polar expeditions, and while it would take fifty years to reach the twelve-mile depth and would require approximately one hundred million dollars to finance, such an exploration might reveal the existence of new chemical elements and would secure accurate information as to temperature conditions. Sir Charles also pointed out that such a venture, if planned and conducted internationally for the achievement of scientific knowledge, would be a bond of union between nations.

Senator Luiggi also expressed his appreciation for the honor accorded him and reviewed the policy of the Italian government in subsidizing public works and public utilities under private initiative. The Italian government subsidizes the reclamation of marsh lands in some cases up to sixty per cent of the cost. Hydroelectric power development is furthered by the payment by the government to public-utility companies of 40 lire per horsepower-year for a period of 50 years for newly developed resources. Senator Luiggi also told of the Apulia aqueduct whose total length, including distribution mains, is nearly two thousand miles and comprises three tunnels of six, seven and eight miles in length. This aqueduct delivers water to one and one-half million people and at several points in its course the flow is utilized in hydroelectric plants, the current from which is used to pump the water to communities above the level of the aqueduct.

## A Pioneer in Engineering Education

THE Centennial Celebration of the Rensselaer Polytechnic Institute, which was held in Troy, New York, on October 3 and 4, 1924, was not only a feast of congratulations on the completion of one hundred years of successful service, but a tribute to the founders of the school for their foresight and courage in visualizing and establishing an institution that started a "tradition of immeasurable consequence." In 1824 an institution to teach the "application of science to the common purpose of life" was a radical departure from the then accepted ideals of education. While such a school had been thought of, it was not until the fortunate combination of benefactor and leader in the persons of Stephen Van Rensselaer and Amos Eaton made possible the definite formulation of an institution on an enduring basis.

The original idea was a school for the farmers on Van Rensselaer's broad estates, but the method of instruction and the leadership were so sound that, as transportation and industry became factors in the life of the nation, graduates were ready to apply their scientific training to the problems of these great agencies of civilization. One hundred years of achievement with graduates contributing their bit in every field of engineering to the advancement of man's well-being is proof of the solid foundation laid by Van Rensselaer and Eaton.

In the light of the present, Amos Eaton must have been a giant of science and education. He is credited with the inception of the Institute, and his influence prevailed during the early years of its life. He was a botanist of note, he has been styled the "father of American geology," but as an educator he was preeminent, for over seven thousand students attended his classes in natural history, and before he went to Troy many famous scientists had commenced their studies under his tutelage. At the Institute he taught the pure sciences and agricultural education and he planned instruction in domestic economy for women. Funds did not permit the execution of these plans, but Eaton did instruct many women in his private capacity, and several of these students who became well-known educators turned to him for instruction and advice.

In the development of the curriculum at Rensselaer, Eaton suffered many bitter attacks from conservative educators, who saw no merit in teaching by experiment. He stood firm, however, and in a crude way achieved the correlation of the natural and social sciences which is common in modern education.

There is not sufficient space nor is this the proper place to relate the entire story of the growth and influence of Rensselaer during the centennial period. It stood alone until 1847, when the Lawrence and Sheffield Scientific Schools were founded. Massachusetts Institute came in 1865 and then in quick succession Worcester, Stevens, and Case. Rensselaer furnished the example and inspiration for these and the many fine schools that have followed. The important thing is that the spirit of service which prompted the founding of Rensselaer has prevailed, and the exercises of celebration extolled that spirit.

Engineers and scientists of all branches and from many countries gathered at Troy on October 3 and 4 to glorify the achievements of the engineering profession and the ideals of engineering education, and to honor the first school of engineering to be established in an English-speaking country. Distinguished representatives of over two hundred institutions of learning and technical societies, and over two thousand Rensselaer alumni formed the audience for the exercises of the two days, which included a group of inspiring addresses and an impressive pageant depicting the principal incidents in the life of Rensselaer.

At the opening session on Friday morning, October 3, which

was preceded by an academic procession from the campus to the spacious state armory, Secretary Hoover brought a message of congratulation from President Coolidge. He also sounded the keynote of the entire celebration when he pointed out that the education of a body of men in pure science and in the humanities, combined at the same time with a knowledge of the mechanic arts as the vital bridge over which science could alone become the implement of man, was the notable contribution of this institution. Frank Pierrepont Graves, president of the University of the State of New York, presented the greetings of the state, and Harry Esmond Clinton, mayor of Troy, responded for the city. The hearty greetings and congratulations of foreign engineers were presented by Sir Charles Langbridge Morgan, president of the Institution of Civil Engineers of Great Britain, Dr. Henri Abraham, past-president of the Society of Electrical Engineers of France, Senator Luigi Luiggi, president of the Society of Civil Engineers of Italy, and Arthur Surveyer, president of the Engineering Institute of Canada. The morning session was completed by an address by Dr. Palmer Chamberlaine Ricketts, president of Rensselaer Polytechnic Institute, who told of the early history of the school.

In the afternoon a memorial tablet was unveiled by Mrs. Elizabeth Van Rensselaer Frazer, lineal descendant of Stephen Van Rensselaer.

A banquet was held in the evening at which the speakers were Dr. Livingston Farrand, president of Cornell University; Dr. Joseph Henry Odell, director of the Service Citizens of Delaware; and Huger Wilkinson Jervey, dean of the Law School, Columbia University. Following the banquet the first three scenes of the pageant were shown.

On Saturday morning a group of American educators and scientists presented addresses before the gathering in the State Armory. The speakers were James Rowland Angell, president of Yale University; Edward Asahel Birge, president of the University of Wisconsin; Samuel Wesley Stratton, president of Massachusetts Institute of Technology; and Albert Abraham Michelson, president of the National Academy of Sciences. They were followed by the presidents of the Four Founder Engineering Societies; Carl Ewald Grunsky, A.S.C.E.; Frederick Rollins Low, A.S.M.E.; William Kelly, A.I.M.E.; and Farley Osgood, A.I.E.E. The closing speech was by Dr. Ray Palmer Baker, professor of English at Rensselaer Polytechnic Institute, who presented a summary of the one hundred years of achievement at Rensselaer. The rest of the day was given over to alumni gatherings and a reception by President and Mrs. Ricketts, and in the evening the final installment of the pageant was shown.

During the celebration Rensselaer honored a number of speakers by conferring upon them the following honorary degrees: Doctor of Engineering was awarded to the four foreign speakers, Sir Charles L. Morgan, Dr. Henri Abraham, Senator Luigi Luiggi, and Mr. Arthur Surveyer. Doctor of Philosophy was given to Dr. James R. Angell, Dr. Edward A. Birge, Dr. Samuel W. Stratton, and Dr. Livingston Farrand. The degree of Doctor of Science was conferred on Dr. Albert A. Michelson, and that of Doctor of Engineering on the presidents of the four American engineering societies.

## The Lehigh Institute of Research

LEHIGH University has established the Lehigh Institute of Research with the object of encouraging and promoting "scientific research and scholarly achievement in other divisions of learning represented in the organization of the University, and in recognition of the need for further and more exact knowledge in science and in the application of science to the affairs of modern life."

The management of this new organization is vested in a board of directors composed of thirteen members of the University faculty, including President Charles Russ Richards.

As a means for encouraging research and for assisting promising men to secure advanced training in research methods, research fellowships will be set up in the newly formed institute. A fellowship of this kind has been founded by the New Jersey Zinc Company.

## The Dayton Air Meet

THE air races in Dayton on October 2, 3, and 4, provided plenty of thrills for thousands of interested spectators. To the seriously interested onlooker, however, the meet had a greater significance than its thrills, and gave indication of a new orientation in the trend of practical flying.

Nothing illustrated so thoroughly the matter-of-fact character of modern aviation as the first event of the meet, the On-to-Dayton race. As in all the commercial events, Army and Navy pilots were rigidly excluded, but the contest was open to all makes and types of aircraft. Competitors had to start from points 200 miles or more distant from Wilbur Wright Field after September 20, and finish before midnight on October 1. The arrival of twenty-one planes in this race, with but one dropping out on the wayside, the high number of points secured by contestants for speed, carrying capacity and economy, and the casual manner in which the contestants made their way to Dayton with stock models and scarcely any changes, were indeed remarkable.

The program of events also included a group of three other commercial races which gave a conclusive demonstration of progress. Whereas during the war and for years afterward a Curtiss "Jenny" with a 100-hp. motor would carry but a pilot and a passenger at a maximum speed of 75 m.p.h., now the same engine in a similar type of modern touring plane carrying three men will attain a speed of 90 m.p.h.

The United States holds the great majority of airplane records—speed, duration, weight carrying, long-distance and non-stop flights. But it is often maintained that in commercial ships we lag seriously behind. The results of the group of races just mentioned and inspection of some of the visiting passenger planes provide an admirable refutation of this view. For example, the Stout passenger plane flew into Wilbur Wright Field from Detroit, a distance of 206 mi., in 2 hr. 10 min. Equipped with a 400-hp. Liberty motor it can carry a pay load of one ton in addition to a four-hour fuel supply and yet attain a speed of 116 m.p.h. A comfortable enclosed cabin for six, a wash room, a baggage compartment, and the ability to smoke within an all-metal structure provide the occupants with all the comfort and convenience found on the crack liners of the London-Paris air route.

Perhaps the most interesting features of the races were the light-plane contests for prizes of more than $8500 donated by the Bicycle Club, the Engineers' Club of Dayton, and the Dayton *Daily News*. The participants in these contests were mostly young aviators and aviation mechanics, who found in the prizes offered an inducement to build low-powered planes, and who achieved most signal results in spite of lack of equipment and facilities.

The outstanding characteristics of the light planes were their high speed with low power, and their high mileage on a gallon of gasoline. Thus the Driggs-Johnson light plane carried off the valuable Dayton *Daily News* Trophy at an average speed of 54.07 m.p.h., with a 4-cylinder motorcycle engine rated at only 19.7 hp. The plane weighed only 510 lb. fully loaded, yet made 70 m.p.h. on its trial flights. In the second light-plane contest, efficiency as determined by dividing the m.p.h. speed by the amount of gasoline consumed was taken into account. Here H. C. Mummert, of Garden City, won with a two-cylinder Harley-Davison motorcycle engine, averaging only 38.24 m.p.h., but consuming scarcely more than a gallon of gasoline in the fifty-mile race. The light planes achieved these wonderful results mainly because they were so carefully streamlined and because of their cantilever monoplane construction, with no external supporting system to introduce air resistance. The biplanes flown in the light-plane contests were not quite so speedy, but they showed wing spans of only 18 ft. or so, as compared with the 27 ft. of the monoplanes, and their overall lengths were proportionately shorter. Such biplanes are capable of being kept in a garage, and with wings folded will travel under their own power to the nearest flying field on an ordinary road. The consensus of pilots' opinion was that these planes, owing to their low landing speed, were very much safer than the more powerful and heavily loaded ships, which approach the ground at the speed of an express train. There is no doubt that the handiness, safety, and low cost of these planes offer considerable possibility of their general use.

The winning of the Pulitzer race by the Verville-Sperry internally

braced thick-wing monoplane of 1923, in competition with the Curtiss racer of 1922 design with thin, externally braced wings, opened up the question as to which of these two types of wing construction might eventually be reckoned upon to be productive of the higher speed. The sad death of Capt. Burt. E. Skeel emphasized the dangers of an aviator's taking a dive to start off on a race. During the dive, with engine full on, the propeller ceases to function as a propulsive screw and takes on the characteristics of a windmill, which certainly does not retard the engine and may even apply an accelerating torque. Under such conditions the engine may race at very high speeds. Examination of the propeller led to the theory that under the tremendous effects of increased centrifugal forces the glued-up wooden propeller had failed, that the engine had gone next under great vibration, and that failure of the wings had followed the destruction of the engine. The accident is likely to lead to the prohibition of diving starts at such races. The accident does not by any means prove, however, that the limit of speed has been reached. It is perfectly possible to use metal propellers guaranteeing against failure, even in steep dives. The introduction of variable camber in the wings, the possibility of retracting the chassis even in the biplanes which do not provide so ready an enclosure for the reception of the landing wheels, and the use of more powerful engines place 300 m.p.h. within reach according to the opinion of well-informed airplane designers. As a matter of fact, the Army had under way a project of a ship to reach 265 m.p.h., which was not undertaken for lack of funds.

On the whole the meet showed that the trend of aeronautical interest is now distinctly toward economical planes and small air "flivvers," rather than toward the attainment of still higher speeds.

A. K.

## The Flight of the *ZR-3*

THE ZR-3 completed its voyage of over 5000 miles when it landed at Lakehurst on October 15 and added its name to the immortal list of pioneer craft which have opened trails of commerce and communication.

Dr. Hugo Eckner, pilot of the ZR-3, and Captain George Washington Steele, U.S.N., who accompanied the ship in its flight went to Washington on October 16. There they were received by President Coolidge and then entertained at luncheon by a group of naval officers. Dr. William F. Durand, President-elect of the A.S.M.E. was present as the representative of the National Advisory Committee for Aeronautics and escorted Dr. Eckner to a meeting of the Committee for a conference.

In the conference Dr. Eckner explained that the combination of weather reports by radio from land stations and ships with extensive observations from the dirigible made it possible for the ZR-3 to avoid storm zones or to skirt them should such conditions prove desirable to the ship. He also minimized the dangers incident to using hydrogen in a dirigible if the crews are thoroughly trained and well disciplined. Under such conditions he considered the danger from hydrogen is of the same order as the danger from gasoline.

Asked for an opinion as to what was needed to improve the reliability and safety of the airship, Dr. Eckner replied that speed was the main essential and that by increasing the present maximum speed of the ZR-3 five meters per second any storm could be successfully avoided. He said that another important question affecting the safety of airships was gas.

## The Engineering Foundation—Ten Years

FOUNDED in 1914, the first gift for endowment, $200,000, for Engineering Foundation was delivered to United Engineering Society as trustee on February 24, 1915. Only the income could be expended. With no expendable funds accumulated and less than $10,000 a year in prospect, no project could be undertaken immediately.

The great war was on. Scientific associations and engineering societies perceived the urgency for comprehensive organization. In June, 1916, under the leadership of the National Academy of Sciences and the Founder Engineering Societies, the National Research Council was created. No funds were in hand. Engineering Foundation met the emergency by proffering its income. From this beginning Engineering Foundation has coöperated continuously with the National Research Council, especially its Division of Engineering and Industrial Research.

During the war, fatigue failures of airplane parts led to a study of the problem of fatigue of metals. Funds for experimentation were not then forthcoming. In 1919 Engineering Foundation appropriated $30,000 for a two-year program prepared by the National Research Council to be carried out in coöperation with the University of Illinois. Industries became interested; the investigation is still in progress. The General Electric Company, Allis-Chalmers Manufacturing Company, Western Electric Company, and the Copper and Brass Research Association have contributed in cash $61,000, besides services and materials. Valuable additions to knowledge and theory have been made. Other important investigations in this country and abroad have been stimulated.

Engineering Foundation is coöperating with research committees of its Founder Societies in investigations of concrete and reinforced-concrete arches, steel columns for buildings and bridges, mining methods, rock-drill steels, properties of steam, bearing metals, lubrication, and strength of gears. Besides the Foundation's appropriations, totaling $15,000, contributions from industries and other sources aggregate more than $100,000.

In response to request from engineers in the far West, the Arch Dam Investigation was organized. The States of California and Oregon, the U. S. Bureau of Reclamation, the City of San Francisco, several power companies, four universities, and a number of engineers in this and other countries are coöperating. A special fund of $100,000 is being raised for construction of a test dam 60 ft. or more in height, upon which extensive experiments during and after construction will be performed. Subscriptions of the companies and the Foundation total to date $50,000. Additional contributions are being solicited. Observations on existing dams and dams under construction are in progress.

Engineering Foundation joined with the National Research Council in the world-wide Marine Piling Investigation, in establishing the Advisory Board on Highway Research and the American Bureau of Welding, and in compiling a directory of research laboratories in U. S. industries.

The Personnel Research Federation has been created by the collaboration of Engineering Foundation with the National Research Council "for furtherance of research activities pertaining to personnel in industry, commerce, education, and government wherever such researches are conducted in the spirit and with the methods of science."

Engineering Foundation has been also a bureau of information. It has contributed to promotion of research by publications, publicity, correspondence, and participation in meetings of various organizations. Notable among its publications have been a Directory of Hydraulic Laboratories in the U. S. and the Research Narratives. Since January, 1921, the Narratives have been mailed in leaflet form to many men prominent in finance, industry, engineering, and education, and reprinted in newspapers of this and other countries even as far away as Australia, and in technical and other periodicals all over America and as far distant as China. The first fifty Narratives were recently republished as a book by the Williams & Wilkins Company, Baltimore, and are being sold by it for the Foundation at the cost of publication and delivery, fifty cents a copy. The book is having a wide sale.

The latest annual report shows that the total income of the Foundation for nine years, to December 31, 1923, was $169,600. Expenditures have been as follows:

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|---|---|
| Aid in establishing National Research Council | $ 10,500 |
| Support of Division of Engineering and Industrial Research | 26,000 |
| Research projects | 57,000 |
| Promotion of research | 17,200 |
| Administrative expenses | 24,300 |
| Total expenditures | $135,000 |

There was accumulated for commitments and unforeseeable opportunities or emergencies, a reserve of $34,600.

# Synopses of Annual Meeting Papers

## Intakes for Power Plants

By ROBERT W. ANGUS
Professor of Mechanical Engineering, University of Toronto, Toronto, Canada. Mem. A.S.M.E.

THE purpose of this paper is to set forth the desirable features of intakes placed in rivers or other bodies of water for supplying water for hydraulic turbines and other equipment. The paper makes special reference to protection against ice and other large floating substances. It includes a description and discussion and conclusions with reference to an extensive set of experiments made by the author at Niagara Falls, Canada, on a large model of the Niagara River, their immediate object being the design of a suitable intake for the Queenston hydroelectric plant.

## Production Control

By G. BABCOCK
Mfg. Executive, Holt Manufacturing Co., Peoria, Ill.

THE conclusions arrived at by Taylor in his paper on Shop Management in 1903, together with the methods he described, were based upon the thesis that every organized effort of human endeavor can be analyzed into its fundamental elements, and that these elements can be forecast and arranged in an orderly sequence that represents the best combination to attain the desired result. The analysis and arrangement of the elements of production brought about the two great divisions of productive effort that are characteristic of modern industrial management and production control—planning and performance.

In this paper the author presents an outline of the subject of production control in manufacturing, taking up for consideration respectively actual output with given equipment; preplanning; the establishment of manufacturing programs; determination of lot sizes; establishment of the production schedule; operation analysis; stores systems; despatching of work; inspection; maintenance; and forms.

## The Increase in Thermal Efficiency Due to Resuperheating in Steam Turbines

By W. E. BLOWNEY AND G. B. WARREN
Turbine Engineering Department, General Electric Co., Schenectady, N. Y.

EXPERIENCE has shown that as the ratio of the number of stages of a turbine operating in the superheated region to the number operating in the moisture region is increased, the actual thermal efficiency is improved at a rate far in excess of the theoretical value. The results of a large number of tests on turbines at different superheats have been analyzed to form a basis for calculating the gain in thermal efficiency due to resuperheating. In this paper these results are given, showing that the heat consumption of a turbine installation may be decreased from 6 to 7 per cent as a result of resuperheating the steam. There seems to be a rather broad pressure range at which the resuperheating may take place in order to show approximately the maximum saving. The probable gain due to more than one resuperheating has been obtained. These results indicate that, when taking into account the pressure drop in the resuperheaters, the gain possible by going to more than two stages of resuperheating would be very small. The effect of resuperheating upon the exhaust conditions and capacity of the turbine is discussed. Curves are presented to show that the gains due to the resuperheating and the regenerative cycles are very nearly independent of each other, and that these features may be used in the same turbine with the expectation that the savings due to both will be nearly additive.

*THE papers, abstracts of which are published on this and the following pages, are being printed in full in pamphlet form for the Annual Meeting. They may be secured by filling out the blank on page 104 of the advertising section and mailing it to the office of the Society.*

*The balance of the papers to be preprinted for the meeting will be distributed in a special Mid-November issue of MECHANICAL ENGINEERING which will be mailed on November 15.*

*The complete program for the Annual Meeting appeared in the A.S.M.E. News for October 22.*

## Laws Governing Difference between Observed and Theoretical Temperature at which a Liquid Evaporates into Another Gas

By W. H. CARRIER AND D. C. LINDSAY
Respectively President (Mem. A.S.M.E.) and Physicist, Carrier Engineering Corporation, Newark, N. J.

THIS paper contains a report of experiments carried on at Cornell and at the Case School of Applied Science to show that under ideal conditions permitting true adiabatic saturation the wet-bulb temperature is actually the temperature of equilibrium represented in the heat-equilibrium equation, and also to show how and to what extent the wet-bulb temperature observed under ordinary conditions varies from the theoretical equilibrium temperature.

Appendices deal with the process of adiabatic saturation, an approximation of the effect of radiation in raising the wet-bulb temperature above the theoretical, a description of the adiabatic saturator and the experimental methods, and a description of the apparatus and experimental methods used in determining the variation of error in the wet-bulb temperature with variation in air velocity and wet-bulb temperature.

## The Strength and Proportions of Wheels, Wheel Centers, and Hubs

By R. EKSERGIAN
Engineer, Baldwin Locomotive Works, Philadelphia, Pa. Mem. A.S.M.E.

THE paper outlines an approximate analysis of the strength of wheel centers, particularly as applied to spoke wheels subjected to heavy lateral loads such as locomotive driving wheels. To generalize the work, the subject is extended to a study of other types of wheels with a view to approximating their strength characteristics. Inertia loadings and proportions of flywheels are considered briefly.

After a discussion of classification and loading, the author analyzes the characteristics of bending loadings and stresses, considering first the transmission of torque for wheels with heavy and with light rims, second, direct loading in the plane of the wheel, and third, lateral bending in wheel centers. The design of hubs subjected to a pressure fit is then analyzed, and the additional stresses in a locomotive driving wheel due to the offset load on the crankpin are considered. This is followed by a mathematical discussion of the effect of the shrinkage of tires on the compressive stresses in wheel centers. The counterbalance of locomotive wheel centers follows, and the paper concludes with a consideration of inertia stresses and loadings of flywheels and especially of rotors for a.c. generators with rotating fields, and of the inertia proportions of flywheels.

## Design, Manufacture, and Production Control of a Standard Machine

By RALPH E. FLANDERS
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THIS paper describes the methods by which difficulties in manufacture and production control were avoided by the Jones and Lamson Machine Company. The company having passed through a period of increasing speeds and feeds, improvements in methods of doing work and controlling it with satisfac-

tory results as to total machinery time and direct labor cost, directed its attention to overhead, which had suffered a considerable increase due to foremen, clerical work, and cost and production offices necessary for the methods which had been adopted.

Reorganization commenced with a segregation of the products in manufacture so that separate manufacturing organizations and equipments were provided for each product, while a fourth organization was established for repair and special work. A redesign of the product was then undertaken to eliminate as many parts as possible and to standardize parts to fit several types of machines. The shop was arranged on a basis of departments by products. The author describes the turret-lathe shop, the chief manufacturing processes involved, and the routing and stock-room control.

Under the present system the foreman has full power within his territory and can measure the efficiency of his department by the schedule of hours for a given rate of output. There is little need for cost accounting, as the cost of machines is determined on overall operation of the shop, costs of material, labor, overhead and fixed changes being divided by output. An hourly rate basis is used for wage payment.

## The Effect of Inaccuracy of Spacing on the Strength of Gear Teeth

By LLOYD J. FRANKLIN and CHARLES H. SMITH

IN A PAPER presented before the Society in 1912, Professor Guido H. Marx reported results of an extended series of tests to determine the strength of gear teeth at pitch velocities from 0 to 500 ft. per min. During the discussion of this paper it was suggested that further tests be made in order to obtain definite data as to the effect of inaccuracy of spacing on the strength of the teeth at high speeds. At the instance of Professor Marx the authors undertook such a series of tests, the results obtained and a description of the apparatus and procedure employed being given in the present paper.

Among other things the authors found that, in a broad way, at pitch velocities of 1000 ft. per min. and upward, gears whose inaccuracies of spacing do not exceed 0.001 in. will carry twice the load of those having inaccuracies of spacing of 0.006 in.; and that the strength of gears having inaccuracies of spacing of the order of 0.002 in. is about half way between the two. An error of 0.006 in. in the size of teeth tested is much more than will ordinarily be found in first-class commercial cut gears.

## Solid-Injection Oil Engines

By R. HILDEBRAND

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A PAPER dealing with the developments during the last two years in connection with a four-stroke cycle, three-cylinder, heavy-oil engine of the solid-injection type. Combustion chambers and the effect of turbulence are discussed and the design of combustion chamber finally adopted by the author is described. The paper also describes an impact fuel pump and the spray valve developed for the engine, and concludes with some remarks on operation and the future of the solid-injection engine.

## A Graphical Study of Journal Lubrication (Part II)

By H. A. S. HOWARTH

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THIS paper continues the investigations of journal lubrication originally reported to the Society a year ago in a paper of the same title covering the case of a journal completely surrounded by its bearing. (Published in MECHANICAL ENGINEERING, February, 1924.) The case of a partial bearing is considered, and the method of analysis employed leads to the formation of charts for the study of partial-bearing design problems. Some typical examples are solved with the aid of these charts.

## High-Pressure-Bearing Research

By LOUIS ILLMER

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THIS paper presents a synopsis of some research investigations into the laws of friction and deduces a practical method for determining friction coefficients as based upon a wide range of high-pressure-bearing practice. The author's previously published study relating to perfect oil-borne lubrication is briefly reviewed, and it is now found that the identical underlying principles lead to a solution of friction problems where the oil film has been partially or wholly broken down under extreme pressure.

The paper is more particularly directed toward the intermediate field of friction that lies between perfect film and dry metallic friction. The analysis arrives at basic values for the corresponding friction coefficients and introduces a number of modifying constants for use under any given set of operating conditions.

The various kinds of bearing practice have been coördinated to secure results which will harmonize with expectations in practice. Suitable values are presented for determining the coefficient of friction for oil-borne journals; it is shown that the transition from a perfect film to a partial oil-film lubrication causes the friction coefficient to undergo a gradual change rather than an abrupt increase, depending primarily upon the ratio of the actual working pressure to the critical pressure. The friction coefficient applying to high working pressures is also found to vary inversely as the rubbing velocity and the temperature assumed by the bearing.

The oil supply, attendance, location, and structural conditions affect friction losses. Coefficients are furthermore dependent upon the method of working the lubricant between the rubbing surfaces, since parallel flat faces do not have the same capacity for building up an oil film that is possessed by rotating journals. Certain factors have therefore been introduced to make due allowances for such differences in the bearing type.

The matter of pressure limits for slow-moving bearings has also been investigated, and it is shown that the ultimate load capacity is partly fixed by the composition and the abrasive limits of the materials in rubbing contact.

## Review of Recent Applications of Powdered Coal to Steam Boilers

By HENRY KREISINGER

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THIS paper gives a brief statement of the trend of the development for the past two years of the application of powdered coal as a fuel for making steam. It includes treatment of developments in furnaces, driers, and mills. Test results are given from boilers and mills in six central stations using pulverized coal as fuel. The author also discusses mill capacities for various grades of coal.

## An Investigation of the Critical Bearing Pressures Causing Rupture in Lubricating-Oil Films

By LEONARD NOEL LINSLEY

Lieutenant Commander, U. S. N., U. S. S. *Wright*.

THE problem involved in this paper is that of determining the critical or breakdown pressure in the oil films formed in a bearing, using straight mineral oils and also of determining the influence on this critical pressure of the admixture of definite amounts of oleic acid. The author first presents a résumé of the work of other investigators. He describes the apparatus with which his own experiments were conducted and the characteristics of the oils used in his tests. Tests were run with cast-iron and with babbitt bearings, and the results of the final trials, which were run with two of the oils, first untreated and later treated with oleic acid, are given in tabular and graphic form. Efforts to determine the breakdown pressure electrically and the method of measuring film thickness are described. The paper concludes

with suggestions for the extension of the experimental work and the redesign of the test apparatus.

In discussing his conclusions the author remarks that the actual breakdown pressures of oils are several hundred times greater than have hitherto been accepted as possible. Although not definitely established, a relationship between breakdown pressure, speed and viscosity was shown to exist. The actual breakdown pressures at which the film is completely dislodged and the bearing seizes the journal should not be considered as the critical pressure but rather the point at which metallic contact first makes itself manifest. With reference to oils treated with oleic acid the inference cannot be avoided that oils, having the same absolute viscosity but differing in chemical consistency do not have the same resistance to rupture and hence, from this point, differ in lubricating value.

## The Double-Acting Oil Engine

By CHARLES E. LUCKE
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EFFORTS to build large oil engines have been growing in number and in variety of means selected to meet marine requirements, stimulated by the success of motorships and the desire to increase their size or utility in prevailing sizes. Possible stationary applications of larger oil engines have also contributed additional motives and other solutions. Much of this work has not been published and is therefore not as well known to the engineering profession as it deserves by reason of its high quality and the foundation it lays for the future. This paper is concerned with a review of some of this work, with special reference to double-acting engines.

One of the problems considered is that of the possibility of raising the mean pressure in the cylinder to get high horsepower per cubic foot of displacement and without rise of maximum pressure to thereby reduce the weight per horsepower. High injection-air pressures, extra injection air, and supercharging of cylinders are included here in addition to studies of shape of combustion chamber, position of sprayer, and design of spray valve. A second one is that of determining the maximum diameter of cylinder that can be operated at very high or at moderate mean pressures without injury to the metal of cylinder piston or cylinder head, with particular reference to the effect of special designs of these parts for resisting heat damage at a given heat-generation rate or the corresponding relation of diameter and mean indicated pressure.

A third problem is that of securing results in two-cycle cylinders as nearly as possible equal to those of the earlier developed and more widely standard four-cycle, to determine how close the former can be brought to half the weight per horsepower of the latter on the one hand, and on the other for the same size of cylinder and similar engine structures how close to twice the maximum horsepower of a four-cycle engine it may be possible to build the two-cycle. Still another is, from the standpoint of double action of the piston, assuming a given diameter and mean pressure or heat-generation rate established in four- or two-cycle single-acting cylinders, and a construction of cylinder, piston and cylinder head not injured for these values, that of determining the practicability of making both ends develop and the same power in both four-cycle and two-cycle engines. This includes the determination of how close to half the weight per horsepower of the single acting it may be possible to build the double-acting engine or how close to twice the maximum horsepower of the single-acting it may be possible to make the maximum horsepower of the double-acting. Some of the double-acting designs published are reviewed, including the rodless piston and the double opposed pistons.

## Gas Turbines

By LIONEL S. MARKS[1] and M. DANILOV
[1] Professor of Mechanical Engineering, Harvard University, Cambridge, Mass, Mem. A.S.M.E.

THE object of this paper is to present a fairly comprehensive statement of the brake thermal efficiencies that may be obtained from gas turbines of various types. Such efficiencies are functions of the efficiency of the compressor and of the efficiency ratio of the turbine (turbine efficiency). No attempt is made to estimate the compressor and turbine efficiencies which may be realized, now or in the future, but calculations of brake thermal efficiencies have been made for a range of compressor and turbine efficiencies which, it is thought, includes both the present and the future possibilities.

The only published values of the attainable efficiencies of gas turbines, calculated on the basis of the true (variable) specific heats of the gases are contained in the fifth edition of Stodola's Dampf-und-Gas Turbinen, which is not as yet available in the English language. These values are exclusively for turbines of the explosion type and for a certain specified range of conditions. The purpose of this paper is to extend that range and to include also constant-pressure-combustion turbines. In addition there are given the efficiencies for the more promising modifications of the simple cycle, such as (1) regeneration, (2) water injection into the combustion space, (3) steam generation by the exhaust gases for use in a separate steam turbine, and (4) the use of an exhauster for extending the operating pressure range of the gas turbine.

## An Experimental Determination of the Coefficient of Discharge of a Flow-Measuring Nozzle for Air

By A. J. NICHOLAS and J. E. NICHOLAS
Respectively Department of Mechanical Engineering, Pennsylvania State College, State College, Pa. (Jun. A.S.M.E.), and Mechanical Engineer, Eckley, Pa,

THIS paper presents a careful investigation of the characteristics of a standard type of nozzle for measuring air flow, the apparatus and methods of measurement being refined and precise. The nozzle was tested for a complete range of pressure ratios, and the coefficient of discharge, defined as the ratio of actual to theoretically calculated discharges, was found to be practically constant for this range. The coefficient of discharge was also found to be high and constant at low velocities for a small nozzle, contrary to accepted findings. In view of the extensive use of flow nozzles, these findings merit interest and discussion.

## Recent Developments in the Burning of Anthracite Coal

By W. A. SHOUDY and R. C. DENNY
Respectively Supt. of Steam Stations, Adirondack Power and Light Corp., Schenectady, N. Y., and Test and Research Dept., Combustion Engineering Corp., New York

THE anthracite coal referred to is that of which not less than 95 per cent will pass through a $^3/_{16}$-in., round-mesh screen and not 20 per cent through a $^3/_{32}$-in., round-mesh screen. The coal is burned on Coxe stokers.

The paper describes successive furnace designs made at the Amsterdam (N. Y.) steam station of the Adirondack Power and Light Corporation. The boilers are of B. & W. design and of 1345 hp. Each boiler has two Coxe stokers. Attempts were made to improve on the customary furnace design to which the following objections are made: (*a*) Tendency to stratification of gases; (*b*) high carbon content of ash; (*c*) loss of fires to ashpit, and (*d*) lack of flexibility.

The first experimental furnace had, in addition to the ignition arch, a second arch over the rear of the grate. With this arrangement improvement, if any, was slight. A second furnace was then built, with better results but with defects due to slagging. A third furnace, of multiple-arch design, was finally constructed. Tests reported in the paper show that with the final design stratification can be practically eliminated, as well as ignition troubles, even with low-grade coal. A decided improvement has also been made in the burning of undersizes. Higher rating was obtained with low-grade coal (60 per cent undersize and 33 per cent ash) in the multiple-arch furnace (300 per cent) than in the single-arch furnace (190 per cent). The boilers and furnaces are described and illustrated, and the principal data and results of the tests are given in the form of curves.

## The Turbine Designer's Wind Tunnel

BY H. LORING WIRT

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THIS paper describes briefly methods developed by the General Electric Co. to accelerate advancement in the art of turbine design and to insure correct design of all the elements of a turbine that control and guide the flow of steam from the control valve through the nozzles, buckets, and exhaust hood to the condenser. The rapid improvement of the airplane is based on an infinite amount of painstaking wind-tunnel research and in much the same way improvements in the turbine have been pointed out by the "Turbine Air Test," which in reality is the turbine designer's wind tunnel.

Air-test methods similar to those described in the paper could be adapted to ventilating and aerodynamical problems, in fact they are universally applicable to any problem that concerns the flow of a fluid, be it compressible or incompressible. The methods can be used to design streamline valve passageways for water, air, or steam, or to indicate the flow around the nose of a projectile, airplane, propeller, or ship model.

## Mechanical Springs

BY JOSEPH KAYE WOOD

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THE paper treats the subject of mechanical springs collectively in the hope of clarifying theories of design and of assisting in the ultimate standardization of springs. After defining a mechanical spring and a mechanical spring material, it considers the general cases of a unit cube stretched by a tensile force, of replacing the cube by a bar, and of applying the load transversely. From these are established load-deflection-rate formulas for tension and torsion. Formulas for safe maximum load, safe maximum deflection, and safe maximum work are then derived in general terms containing constants which may be determined for stress method, material, form, etc. Under spring requirements of mechanical design, load-deflection characteristics are first considered, followed by those for safe maximum load and deflection and safe maximum work or resiliency. The paper then discusses the adaptability of springs to the requirements of mechanical design and the constants of material, dimension, stress method, and form of section. In the conclusion it is stated that the general or collective method of treating mechanical springs should eliminate much of the complexity and diversity of the subject.

## The Zoelly Steam-Turbine-Condensing Locomotive

BY DR. H. ZOELLY

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THIS paper, by the former chief engineer of the Escher-Wyss Company and designer of the well-known steam turbine bearing his name, gives particulars regarding a 1000-hp. experimental condensing locomotive driven by a six stage impulse turbine and employing a cooler for the condensing water in which air is brought into intimate contact with the water, heated and saturated with vapor, the heat necessary for evaporation being withdrawn from the water to be cooled. Mention is also made of a 2000-hp. locomotive of the same type now being built at the Krupp Works, Essen, which differs from the experimental machine only in a few sundry details; after which theoretical considerations regarding feedwater heating and recooling are discussed at some length. A table is included which presents calculations of the steam and coal consumptions of simple and compound condensing and non-condensing piston locomotives using superheated steam, and of the Krupp turbo-locomotive, all with and without feedwater heating, and shows the turbo-locomotive when employing feedwater heating and preheating of combustion air to have a coal consumption of but one-half that of a simple locomotive provided with a feedwater heater.

## Books Received in the Library

AUTOMOTIVE CONSTRUCTION AND OPERATION. By J. C. Wright and F. C. Smith. John Wiley & Sons, New York, 1924. (Wiley Trade series.) Cloth, 6 × 9 in., 446 pp., illus., diagrams. $3.

Describes the various elements of the automobile, explains their functions and methods of operation, and gives directions for maintaining them in proper condition. Is suited for use as a textbook for students or as a reference book by owners and salesmen.

CALCULUS FOR ENGINEERS. By Ewart S. Andrews and H. B. Heywood. Second edition. Scott, Greenwood & Son, London, 1924. (Broadway engineering handbooks, v. 13.) Cloth, 5 × 8 in., 269 pp., diagrams. $3.

This textbook, the outcome of the combined efforts of an engineer and a mathematician, is an endeavor to present a course of study which will equip an engineer with a satisfactory knowledge of the subject. Engineering calculus is treated as a subject of engineering, and each new development is approached through an engineering problem. Much consideration is given to typical engineering applications.

DAVISON'S TEXTILE BLUE BOOK....1924/25. Office edition. Davison Publishing Co., New York, 1924. Cloth, 7 × 9 in., 1778 pp., maps. $7.50.

The 1924 edition of this well-known annual has been carefully revised by the removal of all mills that have ceased business and the addition of 579 new manufacturers. The directory covers all branches of the textile industry, includes information on merchants, supply dealers and dyers, as well as on manufacturers, presents the data in various classifications meeting the needs of dealers and salesmen.

DIESEL AND OIL ENGINEERING HAND BOOK....LAND AND MARINE. By Julius Rosbloom. Second edition, enlarged. Technical Publishing Co., Los Angeles, Cal., 1924. Fabrikoid, 5 × 7 in., 780 pp., illus., diagrams, tables. $5.

Contains much information on the construction of land and marine types of Diesel and oil engines, on their operation and on maintenance. The various commercial types are described in some detail. The final chapter has only slight connection with the others. It is devoted to oil-well machinery and to well drilling, pipe lines, etc.

THE ENGINEER. By John Hays Hammond. New edition. Charles Scribner's Sons, New York, 1924. (Vocational series.) Cloth, 4 × 7 in., 218 pp. $1.

The series in which this book appears is intended to assist the youth in choosing a profession. Drawing on his wide experience, Mr. Hammond explains what an engineer has to do, the advantages and disadvantages that the career offers, the training that the engineer should have, and the various branches of the profession that may be selected.

GRAPHIC STATISTICS IN MANAGEMENT. By William H. Smith. McGraw-Hill Book Co., New York, 1924. Cloth, 6 × 9 in., 360 pp., diagrams, charts. $4.

The author's aim has been to coördinate statistical practice and graphic usage in such a way as to give the business man a presentation of the subject which will be understandable from his point of view. The subject is approached through a study of the principles of graphic practice. Next, the principles of statistical activity are discussed and, finally, both practices are considered as they apply to the different phases of business or to the individual business.

RADIOTECHNIK, v. 1; Allgemeine Einführung. By J. Herrmann. Walter de Gruyter & Co., Berlin and Leipzig, 1924. Boards, 4 × 6 in., 128 pp., illus., diagrams. 1.25 gold marks.

The first of a series of brief volumes on radio technology. It furnishes a concise survey of the present position of radio communication, telephonic and telegraphic. Special attention is given to the theoretical principles of the subject. Sending and receiving apparatus are described.